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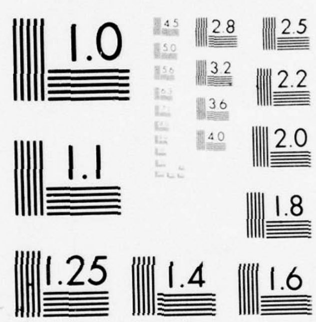

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ANALYSIS OF NUCLEAR AND COAL FUELED TOTAL ENERGY SYSTEM OPTIONS FOR FORT KNOX, KENTUCKY

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30 June 1977

Final Report

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ABSTRACT

A Total Energy System (TES) is designed to supply the thermal and electrical energy requirements of Fort Knox, for Kentucky for a period of 30 years, with startup scheduled for early 1985. Considered for use as the central station power plant for this system are a combined coal gasification, fossil-fired gas turbine (CGGT) power plant and a direct Brayton cycle high-temperature gas-cooled reactor, helium gas turbine (HTGR/GT) power plant. Several utility system configurations affording different thermal/electrical energy demand ratios are studied for each supply option. With the primary system optimization criterion being the choice of the TES providing a minimum of total energy costs over the system lifetime, it is found that the optimal thermal/electrical load split for each supply option occurs at approximately 80% of the base's total energy demands supplied thermally. Within the limits of the unit-cost assumptions made and for the range of cases studied, it is found that the present-worth total cost of the optimized HTGR/GT system (in 1985 dollars) is $234.5 million and the corresponding optimal system cost for the fossil CGGT alternative is $182.2 million.

ACKNOWLEDGEMENTS

This report is the work of the following members of the Nuclear Engineering Department of the Massachusetts Institute of Technology:

Prof. Michael W. Golay, MIT faculty, Project Director,

Mr. Frederick R. Best, MIT student, Analyst and Coauthor,

Mr. Steven B. Goldman, MIT student, Analyst.

This report was typed for publication by Ms. Eva M. Hakala.

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CHAPTER 1
INTRODUCTION

1.1 Foreword

This is the final report under a contract between the Massachusetts Institute of Technology and the United States Army Corps of Engineers to develop a conceptual design for a Total Energy System (TES) supplying both electrical and thermal energy to large U.S. Army bases. The system discussed in this report is a third iteration optimization of the design for a 1985 Total Energy System for Ft. Knox, Kentucky. It is a successor design to that of a similar TES study performed previously for Ft. Bragg, N.C. Use of both nuclear, and fossil-fueled coal gasification-gas turbine power stations are considered as well as the dependence of power station costs upon the thermal/electrical apparatus mix in the customer sector. The sensitivity of TES costs to changes in capital costs, fuel costs, end use equipment costs, the marginal cost of the electrical distribution system and Thermal Utility System (TUS) cost is also presented. Recommendations are made regarding the optimum TES for Ft. Knox. The Ft. Knox simulations have been performed using a significantly improved model, TDIST2 [1] which has been developed recently in this project.

It is found that a minimum cost Total Energy System for both the nuclear and fossil options occurs when the thermal/

electric space conditioning split is set at 80%/20%. Additionally, it is shown that for the fossil-fired plant to remain less expensive than the nuclear option, the projected cost of coal must remain less than $70/ton averaged over plant lifetime. The plant is to be located at Arnold Bottoms, three miles northeast of the population center.

1.2 Background

During the past ten years, oil and natural gas have supplied 75% of the nation's energy needs, with coal supplying 21% and all other energy sources, including nuclear, accounting for only 4% of the total. [2] Oil and gas have been the preferred energy sources because they were easily obtained, transported and converted to electrical and thermal energy. Recently, however, the scarcity of natural gas and the rising cost of foreign, interruptible oil supplies has led to consideration of alternative energy sources for meeting energy demands. Solar power, wind power, geothermal, fusion and many other energy sources are being investigated and developed to meet national energy needs. However, coal and nuclear power are the principal competitors in the current energy market place. Each fuel has its own characteristic advantages and disadvantages, some of which are listed in Table 1.1.

As is seen in Table 1.1, there is no decisive factor which would lead to choosing one energy source over the other. In the report prepared by Metcalfe and Driscoll, "Economic Assessment of Nuclear and Fossil-Fired Energy Systems for

TABLE 1.1

NUCLEAR VERSUS COAL PLANT CHARACTERISTICS

	Nuclear		Coal
1.	Complex licensing procedures and operating requirements.	1.	Can be operated a maintained by fewe and less-well-tra personnel than a nuclear unit.
2.	High capital cost	2.	Lower capital cos
3.	Low fuel cost	3.	High fuel costs
4.	Several years (3-6) of operation on a single fueling	4.	Impractical to st more than a few mo fuel supply on si
5.	Low environmental impacts	5.	Meeting exhaust e standards imposes economic penalties
6.	Low risk, but high consequence reactor safety hazards exist	6.	Can be located c to load center
7.	Requires relative isolation of the plant (exclusion area)	7.	Airborne chemical sions impose sig public health risks
8.	Cooling towers required for dissipation of waste heat	8.	Use of gas turbines allows waste heat exhaust to the at phere
9.	Technology for the disposal of radioactive waste is not established	9.	Successful reclam of stripmine sites very expensive, a some cases not de strated to be poss

DOD Installations," [3] nuclear plants and fossil-fired gas turbine plants are shown to be economically competitive in the size range of interest (50-100 MWe). Metcalfe, et al., considers pressurized water reactors (PWR), high temperature gas cooled reactors (HTGR), conventional coal and oil fired plants, as well as preliminary calculations on coal gasification gas turbine plants (CGGT). Metcalfe's work is used in this report as the source of economic data regarding nuclear power costs.

1.3 Report Outline

In Chapter 2 are developed the model of the coal-gas gas turbine (CGGT) plant used for comparison with a 38%-efficient HTGR Brayton cycle power station. Note that the nuclear analysis is not restricted to use of an HTGR power station. An LWR power plant at reduced efficiency would be able to provide both heat and electric power via a steam-extracting turbine. In this chapter also are outlined the selection of specific components, the sizing of these components and the calculation of fuel consumption rates.

In Chapter 3 are explained the consumer classifications used in the analysis of the thermal and electrical loads of Ft. Knox. Load schedules for each consumer group are presented. The thermal utility system (TUS) piping distribution system is explained in Chapter 4 together with the

design criteria which were used. In Chapter 5 are presented the energy demand simulation results obtained in examining the TUS as described in Chapters 3 and 4. The effect of the consumer thermal-electrical demand mix on TUS loads is also described.

The optimization of the TES with respect to overall cost is discussed in Chapter 6, with Chapter 7 summarizing the report's conclusions and recommendations. Appendices are included to document key technical aspects of the calculations employed to develop the results.

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CHAPTER 2

COAL GASIFICATION FOSSIL-FIRED GAS TURBINE PLANT ANALYSIS

2.1 Introduction

To ensure a valid economic comparison between a High Temperature Gas Cooled Reactor (HTGR) and a fossil fired alternative, the model of the fossil fired plant should be as well developed and understood as the HTGR model. The fossil-fired plant model should represent realistically the available technology, but not be given credit for potential and as yet undeveloped technological improvements. A Coal Gasification-Gas Turbine (CGGT) plant is selected for analysis based on the preliminary economic comparison performed by Metcalfe. [1] This section of the report outlines the development of the plant model, and describes the final CGGT model.

2.2 Selection of Coal Gasification-Gas Turbine Components

Coal Gasification and Gas Turbine reports [2,3] prepared previously in this project, are used as the basis for the selection of components. The objective of the selection process is the specification of a set of mutually compatible components, well suited to the requirements of a Total Energy Utility System. The selection of a coal gasifier, gas purifier, gas turbine and waste heat exchanger is explained in the following sections.

2.2.1 Gasifier Selection

Table 2.1 (reproduced from the project Coal Gasification Report [2]) summarizes the important system parameters of the currently available commercial coal gasification units. The most crucial of these parameters are those which affect component complexity (and thereby reliability), system compatibility and cost. It is seen that the heating value of the gas should not be considered as a controlling parameter in the selection of process equipment, since relatively simple changes in turbine combustors allow wide variations in fuel heating value. Thus, the greatest weight - in selecting a given component - is given to component compatibility within a complete system, and a history of proven successful performance. Realistically, it should be pointed out that no single gasifier is clearly superior to all others, with the result that the selection of any gasifier would imply gasification costs of approximately the same value.

With these considerations in mind, the Lurgi gasifier is chosen for use in the project's CGGT system because of its history of proven technology, simple construction and reliable operation. Additionally, the output pressure of the Lurgi product gas (300 psi) is suitable for compressed gas storage with minimum compressive work, the Lurgi unit can use air rather than oxygen as a gaseous feedstock (obviating the need for an oxygen plant), and required coal preparation

TABLE 2.1

Competing Advantages and Disadvantages of the Available Gasification Processes

Process	Lurgi	Kellogg McDowell-Wellman	Riley-Stoker	Koppers-Totzek
ADVANTAGES	Under continuing development by the Lurgi Co., and General Electric Co. Produces high pressure gas.	Does not require O_2 feed for high-Btu gas production.	Small unit capacity - permits precise capacity specification.	Rapid startup. Does not consume product gas in steam generation. A pressurized product gas design is being developed.
DISADVANTAGES	Difficulty with caking coals - Stirring arms required Long successful field experience	Process not improved in recent years, and little current improvement work underway.	Little field experience accumulated.	More extensive coal preparation required than with other processes. Needs an oxygen feed stream.

operations are minimal. It is notable that several other development groups [4,5] have also selected Lurgi gasifiers as the basis for combined cycle plant designs.

The Lurgi does have at least two minor drawbacks (neither of which warrants changing to another gasifier), the low heating value of the product gas, and difficulty in using caking coals. The low heating value of the product gas principally affects the required gas storage volume. The Lurgi Company has treated the caking problem by adding rotating arms, called stirrers, to agitate the coal bed and has successfully gasified caking coals.

2.2.2 Gas Purification

Table 2.2[2] lists a few of the most attractive purification processes available for removing sulfur from the gas. Most proposed large (1000 MWe) [4,5] combined cycle plants use a series of sulfur removal processes, such as potassium carbonate - to Claus purification - to Scott-tails processing. This sequence is used to reduce the loss rate of the catalyst in the Claus purification process by reducing the volume of gas passing through the Claus system. It is thought for the small sized plant proposed for the Ft. Knox TES (142 MW(t)), that the added cost and complexity of the potassium carbonate system is greater than the corresponding savings in Claus catalyst achieved by using the potassium carbonate system.

TABLE 2.2

H_2S Removal Processes

Process	Sorbent	Final H_2S (ppm)	Operating Temperature (°F)	Final Product	Comments
Purisol	N-methyl-20 pyrrolidone	2	below ambient	H_2S	Expensive sorbent; all oils and tars must be removed; works best at high pressure.
Claus	SO_2	100-500	500-600	S	Requires Stretford treatment of tail gas
Stretford	Na_2CO_3, Na_2VO_5, ADA	1	50-120	S	Expensive sorbent; high liquor rates. Thiosulfate formation minimal below 100°F; proven process

For this reason, a simple Claus purification system with Stretford tails-processing is recommended for the CGGT plant.

2.2.3 Gas Turbine Selection

Table 2.3 is generated from the project Gas Turbine report [3]. The table displays the principal-characteristics of currently available gas turbines which are relevant to a CGGT plant. The Turbo-Power Marine FT4C Power Pac [3] is selected as the basic unit of electrical generation. Initially, the FT4C was selected for use in the project design because of its unique design which decoupled the electrical generator turbine from the compressor-combustor turbine. This feature would permit a large fraction of the combustion gas flow to by-pass the electrical generator, and to supply heat directly to the Waste Heat Exchanger. It was thought that by-pass flow would be a convenient method of shifting the ratio of electrical/thermal power produced, as the TES demand changed through the day. However, the winter peak thermal load at Fort Knox is so much greater than the electrical load that merely using FT4C turbines to supply all the thermal power would require additional turbines, with most turbines operating solely as hot water heaters. The solution to this problem is to use a separate gas-fired water heater. Thermal Power (hot water) is produced by the FT4C exhaust waste heat exchangers (as base-loaded heat

TABLE 2.3. RELATIVE MERITS OF THE COMPETING TURBINES EXAMINED IN THIS STUDY

Criterion / System	System Cost	Low BTU Gas Progress	Thermal Control	Fuel Usage	Maintenance Cost	Overall Reliability
TP&M FT4C Aircraft Derivative	-1	0	+2	0	+2	+2
W-501 Westinghouse Industrial Turbine	+1	+1	0	+2	-1	0
W-251 Westinghouse Industrial Turbine	0	+1	0	-1	-1	+2

Key: +2 - Superior
+1 - Good
0 - Fair
-1 - Poor

sources), and also by the gas-fired water heater when necessary.

The question then is, since a gas-fired water heater is being used, why not simply pipe gas to the load points and use conventional heating systems? The answer is made up of two parts.

1) Use of a central station gas-fired water heater (together with the turbine-exhaust water-heaters) reduces fuel consumption and therefore fuel costs. This results in a 90% saving in fuel costs (see Appendix A.1),

2) The design concept of the CCGT model is based on a one-for-one replacement of any proposed HTGR/GT plant, powering the Ft. Knox TES.

For those two reasons, the central station CGGT concept is retained. The FT4C turbine is selected as the turbine unit of choice because its combustor can be easily modified for use of low BTU gas, its unit size (26.3 MWe) is easily matched to the Ft. Knox load, and the capital and operating/maintenance costs of the FT4C are reported by utilities [3] as being among the lowest of the available units in the capacity range of interest. It is felt that for increased availability there should be four gas turbine generators, three running and one a backup unit.

2.2.4 Thermal Energy Storage

The thermal load of Ft. Knox varies typically on a daily cycle as shown by Figure 5.7. There are three ways by which this thermal demand can be supplied:

1. Produce thermal power at the required average daily rate; and use a thermal reservoir to store energy when thermal demand is low, and to release heat when thermal demand is high,
2. Produce thermal energy at the instantaneous rate required by the Thermal Utility System (TUS) load, and
3. some combination of options 1 and 2.

Option 1, thermal energy storage, is the most economical approach for a TES using an HTGR power station, because this option minimizes the size and cost of the HTGR. Since the HTGR is by far the most expensive item in the system, minimizing HTGR cost, as a first approximation minimizes overall system cost.

However, Option 2 could be more attractive for the CGGT system than Option 1. Utilizing Option 2 instead of Option 1 for a CGGT system affects only the designs of the gas fired-water heater, the thermal reservoir, and the gas storage tanks. Implementing Option 2 for a CGGT system requires increasing the size of the gas storage tank(s) so that they can store sufficient gas to permit absorption of the thermal load swings. Option 2 also requires a larger gas fired water heater (sized to meet peak demands), but it eliminates the

need for a thermal reservoir. An economic balance must be struck between increasing costs due to increasing gas tank storage volume and water heater size, compared to decreasing costs due to eliminating the thermal reservoir. As is shown in Appendix A.2, it is much less expensive (on a specific energy cost basis) to store energy as hot water than as gas. Therefore, Options 2 and 3 are not considered further in the economic evaluation of possible designs.

Hot water may be stored in steel tanks, pre-stressed concrete vessels, excavated rock caverns or high pressure aquifers. Steel tanks are selected as the storage mechanism, because they have a proven operating history and (for the size range of interest) they may be shop-fabricated. Rock cavern or aquifer storage depends on site geology, and since this information was not available (and in any case would vary from site to site) these techniques are not considered further.

2.2.5 Gas-Fired Water Heater

Gas-fired water heaters of the required capacity are readily available from several vendors. [6] Two water heaters are used in the CGGT plant to improve system availability. Each gas-fired water heater supplies approximately 25% of the winter peak thermal load, the rest of the thermal energy is recovered from the gas turbine exhaust waste heat exchangers.

2.3 Component Sizing

The size or number of the various components in the CGGT system is set by the loads which these components must serve.

2.3.1 Gas Turbine Sizing

The peak electrical demand of the optimal TES for Ft. Knox is 54 MWe. Three TPM FT4C (each 26.3 MWe) turbine generators are considered to be used to supply this load. Three small units are used (rather than a single larger one) in order to insure a high system availability. Although the FT4C is rated at 26.3 MWe, it has a reserve capability of 31.1 MWe such that in an emergency one FT4C can supply 60% of the peak electrical demand. An additional turbine is included in the plant design (giving a total of four turbines) to act as a backup unit.

Each FT4C has an exhaust waste heat exchanger, which recovers a maximum of 32 MW(t) from the hot exhaust gases. This thermal energy serves the TUS.

2.3.2 Lurgi Gasifier System Sizing

The smallest commercially available gasifer unit has a capacity of 1.87×10^8 BTU of gas per hour. The design day requires use of four Lurgi units. Forced-outage backup capacity for the Lurgi units is considered to be accounted for by their inherently high availability, and one additional unit. The final design utilizes five independent

Lurgi units.

2.3.3 Sizing the Thermal Reservoir

The largest variation between peak thermal demand and thermal output schedules occurs on the design winter day as shown in Fig. 5.27. The energy mis-match between the thermal demand and thermal supply schedules determines the energy storage requirements, and, hence, thermal reservoir size. Integrating the energy schedules mis-match over time (the crosshatched area) shown in Fig. 5.27, results in a required energy storage of 362 MW-hr. Using a reservoir water temperature change from 380°F to 150°F, the energy mismatch can be compensated for in a 86,000 ft^3 reservoir. This corresponds to a tank 47.8 ft in diameter and height. The actual thermal reservoir plant design would probably consist of a set of four smaller storage tanks, each tank 20 ft in diameter and 70 ft high.

2.4 Fuel Consumption

A given space conditioning demand can be supplied by several methods,

1. burning of gas at the load point to supply heat (In Appendix A.1 it is shown that this is very wasteful of energy and money),
2. burning of gas at a central station to produce high temperature water (HTW) to supply TUS loads (more economical than option 1),

3. recovery of thermal energy from the electrical generator turbine exhaust, producing high temperature water (HTW) to supply thermal loads; burning of extra gas as required to meet thermal loads greater than the energy available from waste heat exchangers (more economical than either options 1 or 2), and
4. supplying the thermal demand by a combination of electrically-operated heat pumps, HTW heated by turbine exhaust gases, and extra gas burning.

The most economical allocation of electrical space-conditioning and HTW space-conditioning demand is found by determining the thermal loads for various values of electrical/HTW load splits, and then calculating the cost of the corresponding TES. It is found that the total cost of a TES, whether nuclear or coal-fired, passes through a minimum at a thermal to electric split of approximately 80%. Details of fuel consumption and system optimization are explained in Chapter 6. The effect of ambient air temperature variations upon central station efficiency is not considered in these calculations due to the relatively mild climate of the Ft. Knox area; and thus, the small effect of weather upon plant efficiency.

2.5 CGGT Plant Layout

The size and number of components described in Sections 2.2 through 2.3 are shown in a proposed plant plan in

Fig. 2.1 and a schematic diagram in Fig. 2.2. This layout is not completely optimized, but it does incorporate some features designed to reduce costs and to enhance operational convenience and costs. For example, the gas turbines are located close to the gasifers and thermal reservoirs. This reduces the gas pipe run from the gasifers to the turbines, as well as the steam or water lines which run from the waste heat exchangers to the gasifier plant and thermal reservoirs. The gas turbines are arranged so that their exhaust plumes rise in a common area, enhancing overall plume rise.

Because the turbine exhaust waste heat is used to produce hot water for the TUS and is not used in a steam bottoming cycle, there is no need for cooling towers or steam-cycle heat rejection equipment. The plant layout occupies a total area of 73,000 ft^2.

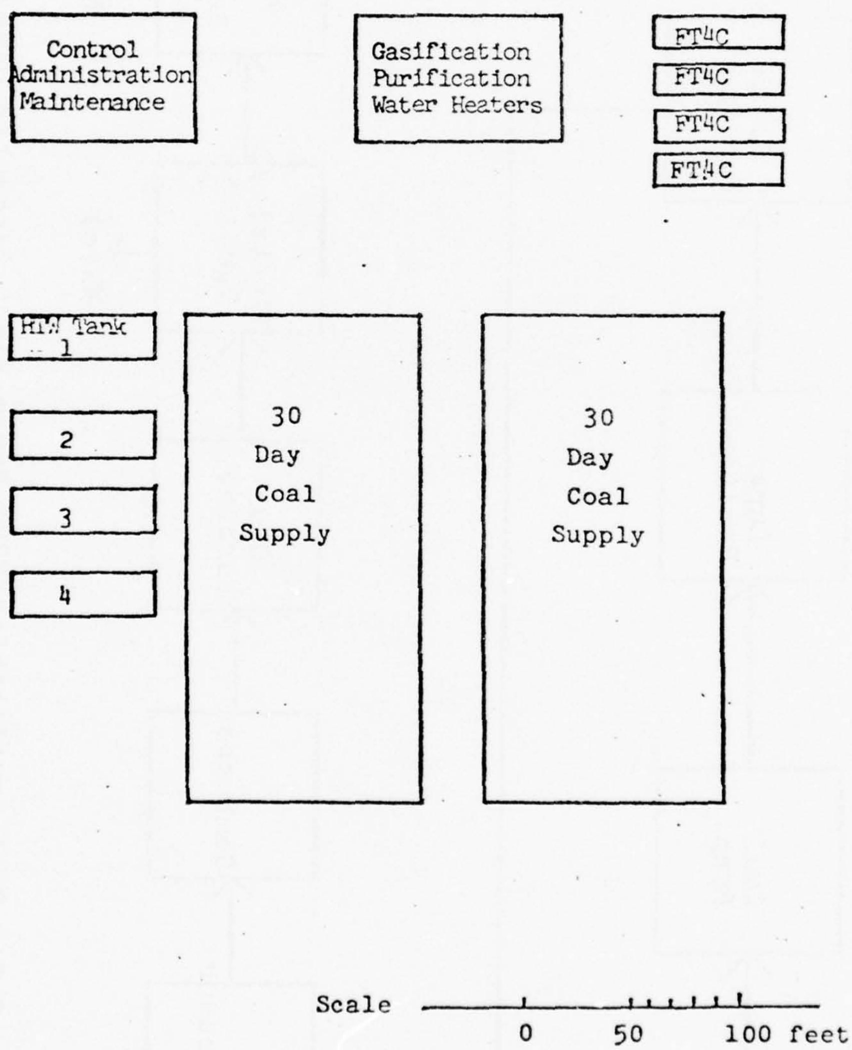

Figure 2.1. Plan View of CGGT Power Station Layout

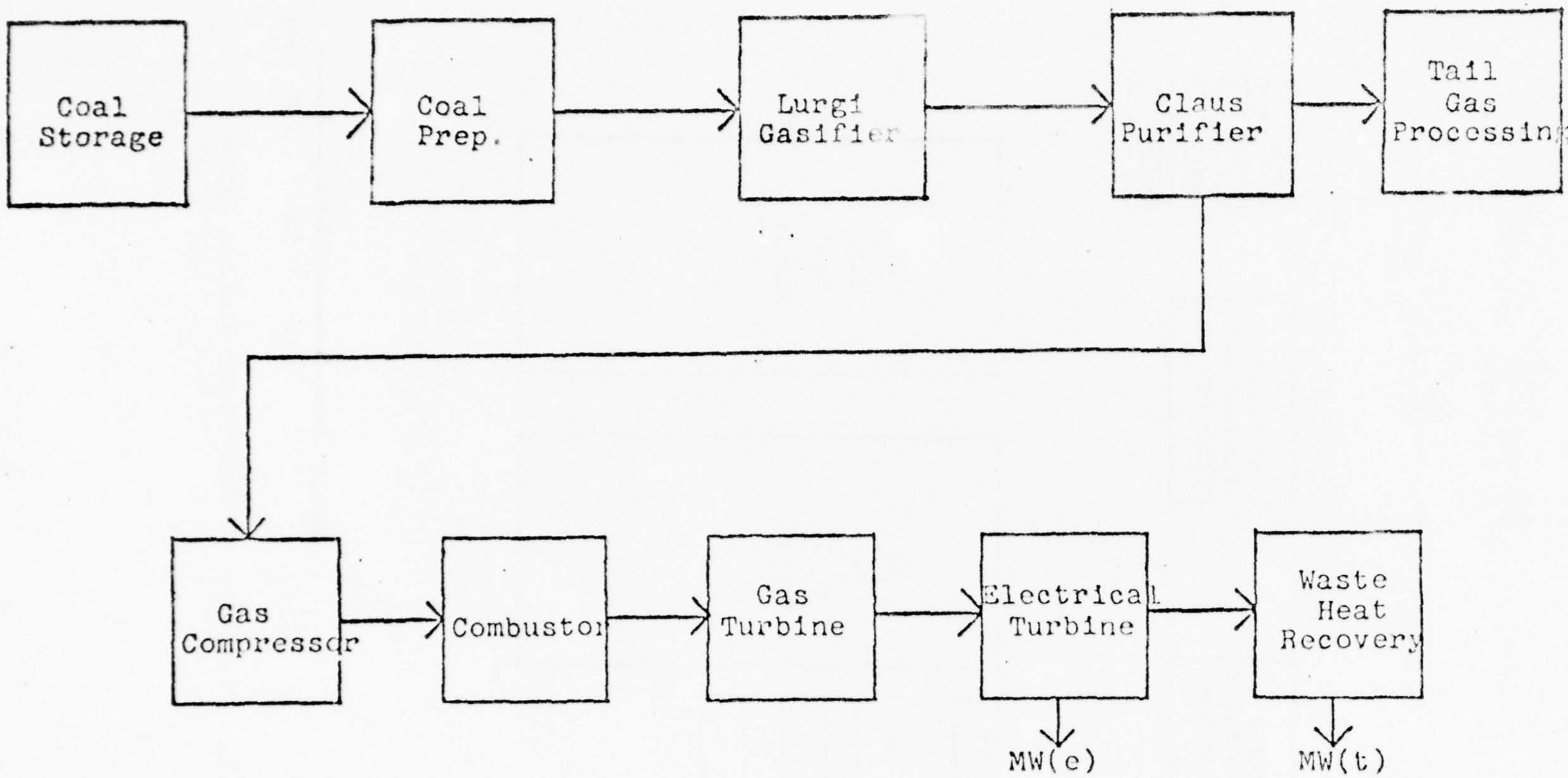

Figure 2.2. Coal Gasification/FFGT Power Plant Schematic Diagram

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CHAPTER 3

FORT KNOX ENERGY CONSUMER MODELS

The nominal startup date for the proposed Fort Knox Total Energy System (TES) is considered to be 1985. To insure that the models of the base's energy consumers accurately reflect anticipated conditions at that future date, the Fort Knox Master Plan for Future Development has been consulted to identify the building types and base configuration to be used in the system analysis. Following extensive discussions with personnel at the U.S. Army Facilities Engineering Support Agency (FESA) at Fort Belvoir, Virginia, and after a field trip to Ft. Knox, it has been concluded that the buildings at Fort Knox may be aggregated into a total of fifteen general energy consumption categories based upon documented building usage and construction characteristics. Table 3.1 lists these fifteen classes with brief descriptions of the "typical" units chosen to represent each category and the number of each found on the base. Appendix B contains more complete descriptions of these buildings, including their construction and usage specifications supplied as input data to the TDIST consumer modelling subroutines.

The magnitudes of the total energy demands of these consumers on the peak winter heating and peak summer cooling days determine the design criteria to be met by the components of the thermal utility system and, depending upon

Table 3.1

Fort Knox Building Category Descriptions

Type 1. Family Housing Modern: Modern family housing includes housing units of two-story, all-brick construction dating to roughly the 1950's. The basic unit accommodates six families and has a total floor area of 6000 ft^2.

Type 2. Family Housing Modern: This representative two-family unit is of brick and wood construction, is entirely on one floor, and has a total floor area of 1500 ft^2.

Type 3. Family Housing Modern: Row: The four-family modern housing units consist of a mixture of two-story brick and combined brick-and-frame construction units, which in some cases are physically attached to form larger connected housing groups. The total floor area of the unit is 7500 ft^2.

Type 4. Family Housing: This family housing consists of large brick residences for high ranking officer single families and high ranking enlisted two family groups. A representative floor area is taken to be 4147 ft^2.

Table 3.1 (Continued)

Type 5. Ft. Knox Large Hospital: Since the hospitals are large loads with unique load characteristics, separate building categories are allocated to them. The large hospital is a new building of concrete construction. It is considered to contain 100 beds in a total floor area of 37,800 ft^2.

Type 6. Ft. Knox Small Hospital: The small hospital is an old building of brick construction. It is considered to contain 50 beds in a total floor area of 27,000 ft^2.

Type 7. Community: Perhaps the widest range of diverse building construction and usage pattern categories is included in this class. Facilities range from recreation buildings to retail sales establishments, units which individually contribute little to the base demand but which in total represent a significant load. The representative unit is assumed to have a floor area of 20,486 ft^2.

Type 8. Administration and Training: The age and construction of these buildings also varies considerably from unit to unit, with the typical structure being formed of a reinforced concrete foundation,

Table 3.1 (Continued)

Type 8. (cont.)	brick walls, and a built-up roof. The representative unit is three stories tall with a total floor area of 24,114 ft^2.
Type 9.	Operations and Maintenance: A machine shop has been chosen to be representative of a wide variety of maintenance buildings. General construction includes either block-and-steel or brick-and-block walls, a reinforced concrete foundation, and a built-up roof. The average floor area is assumed to be 41,850 ft^2.
Type 10.	Troop Housing: Brick: These barracks units are relatively modern three-story dwellings with a capacity of roughly 200 men each. Construction is of brick, and a representative unit has a floor area of 50,959 ft^2.
Type 11.	Troop Housing: Block: Similar in size to the brick units described above, the block barracks consist of older renovated units with an attached mess. Construction is of reinforced concrete and blocks with an average floor area of 51,000 ft^2.

Table 3.1 (Continued)

Type 12. Family Housing: Wood: This representative two-family unit is similar to the Type 2 discussed above, but is entirely of wood construction. The unit has a floor area of 2400 ft^2.

Type 13. Family Housing: Stucco: These small single family, single story stucco dwellings are predominately for junior officers. The unit has a floor area of about 1200 ft^2.

Type 14. Storage: Although many unrelated storage facilities exist at Ft. Knox, they have been combined into a single class due to their similarity of use and relatively small contribution to the total base load. The representative unit is chosen to be typical of a large warehouse with a floor area of 11,421 ft^2.

Type 15. Family Housing: Multiplex: This type is designed to represent an aggregate of two Type 1 units. The essential difference is that for twice the floor area (12,000 ft^2 vs. 6000 ft^2) of the Type 1, the Type 15's have less than twice the exterior wall area (5220 ft^2 vs. 3080 ft^2). This modeling gives a more accurate space conditioning load calculation than would be achieved by merely doubling the Type 1 building loads.

how these demands are supplied, set the required power plant installed capacity and its rated thermal-to-electical energy output ratio. Similarly, the variations in the thermal loads on these days dictate the installed system thermal energy storage capacity required to smooth the imbalances between the diurnal thermal and electrical energy demand schedules. The choice of these design days is thus critica to the ultimate design, configuration and cost of the TES; the weather conditions must be severe enough to insure that the system is capable of meeting the maximum annual power demands, but they must not be so extreme as to cause the system to be grossly over-designed and much more costly than necessary. As specified in Department of Defense "Construction Criteria Manual," DOD 4770.1-M, Oct. 1972, "Engineering Weather Data" AFM 88-8, Ch. 6 is used as the source of design weather data for TDIST2 [1] simulations.

For simplicity, and because coincident wind velocity data was not readily available during the system design period, a constant wind velocity of 5.68 mph from the west has been assumed throughout both design days. The nominal peak solar radiation intensity at Fort Knox for the winter day was assumed to be 390 BTU/hr per square foot of horizontal surface area; the summer day peak was 344 BTU/hr per square foot. [2] Cloudless skies have been assumed, but normal seasonal atmospheric haze and diffusion effects

TABLE 3.2

DESIGN DAY AIR TEMPERATURES

Time	Winter Day, °F	Summer Day, °F
12	11.4	81
1	11.0	80
2	11.4	80
3	12.5	80
4 AM	14.2	81
5	16.5	81
6	19.2	83
7	22.0	84
8	24.8	85
9	27.5	86
10	29.8	88
11	31.5	89
12	32.6	89
1	33.0	90
2	32.6	90
3	31.5	90
4 PM	29.8	89
5	27.5	89
6	24.8	88
7	22.0	86
8	19.2	85
9	16.5	84
10	14.2	83
11	12.5	81
12	11.4	81

TABLE 3.3

SOLAR INCIDENCE FACTORS

(from Ref. 2)

Date	Solar Intensity (BTU/hr.sq.ft.)	Solar Declination[(1)] (Degrees)	Atmospheric Extinction Coefficient[(2)]	Sky Diffusion Factor[(3)]
Jan. 21	390	-20.0	0.142	0.058
Feb. 21	385	-10.8	0.144	0.060
Mar. 21	376	0.0	0.156	0.071
Apr. 21	360	11.6	0.180	0.097
May 21	350	20.0	0.196	0.121
June 21	345	23.45	0.205	0.134
July 21	344	20.6	0.207	0.136
Aug. 21	351	12.3	0.201	0.122
Sept. 21	365	0.0	0.177	0.092
Oct. 21	378	-10.5	0.160	0.073
Nov. 21	387	-19.8	0.149	0.063
Dec. 21	391	-23.45	0.142	0.057

(1) Used in combination with latitude of the location to determine solar position.

(2) Used in combination with solar position to determine attenuation of normal incident radiation due to atmospheric absorption.

(3) Used in combination with solar position to determine attenuation of normal incident radiation due to atmospheric diffusion.

are included as modifying these direct solar radiation intensities. Table 3.3 lists the solar incidence factors used. Summer day building usage and occupancy characteristics have been shifted in time by one hour to account for the effects of Daylight Savings Time, but all load calculations and results are presented in real solar time to allow direct comparison among load profiles at different times of the year.

Figures 3.1-3.15 present the design day space conditioning energy demand schedules computed for each of the fifteen consumer categories. It should be noted that these schedules represent only the net hourly energy gains or losses from the buildings. The corresponding demands to be met by the thermal and electrical energy distribution networks will, of course, depend upon the types and efficiency of the space conditioning equipment used to supply these requirements.

The shapes of the load schedules illustrate the relative effects of the major components of the space conditioning demands. The winter minimum and summer maximum occurring during the daylight hours are due principally to solar radiational heating. These solar effects are compounded, especially for the commercial and public-use building categories, by heat generated internally from lighting and equipment usage. (In fact, for the Administration and Training class, Fig. 3.8, the combined effects of solar and internal heating between noon and 1 P.M. on the winter

Figure 3.1

(A)

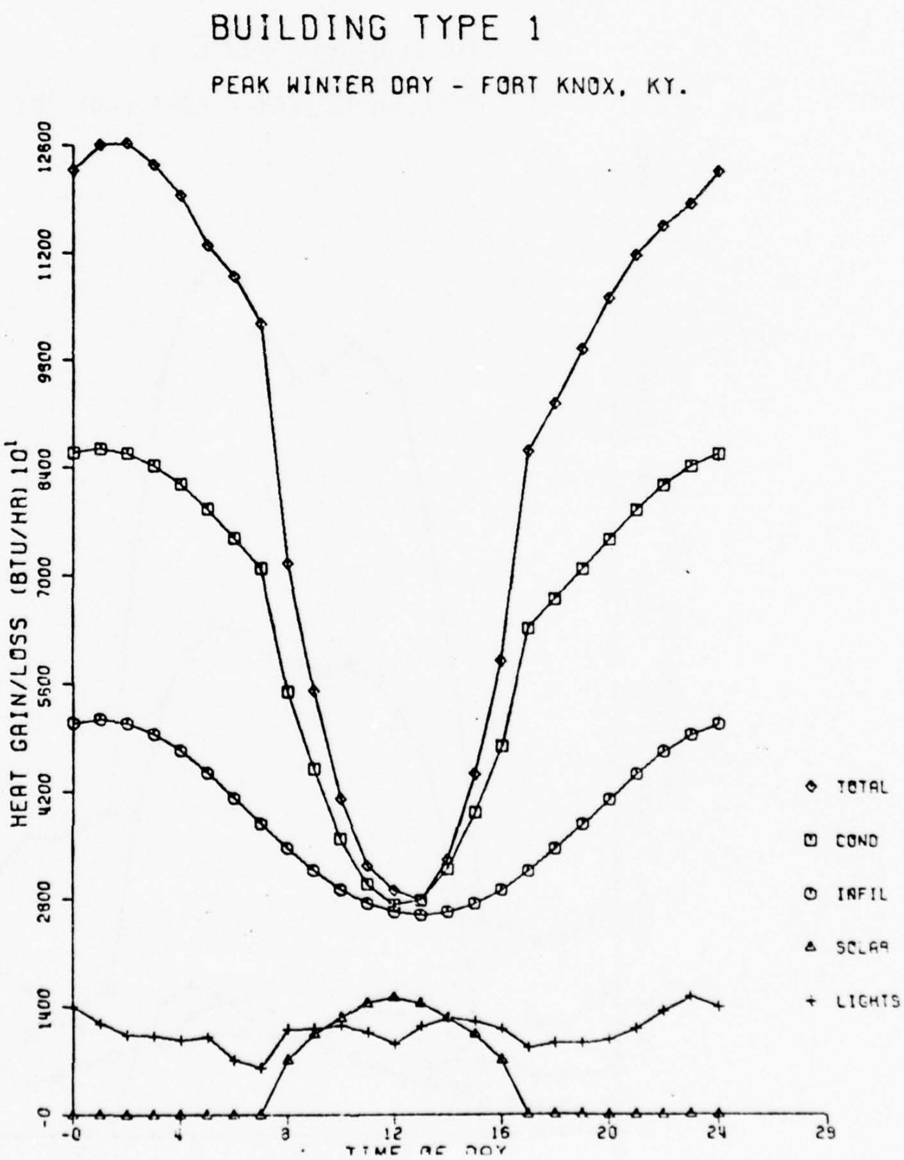

BUILDING TYPE 1
PEAK WINTER DAY - FORT KNOX, KY.
HEAT GAIN/LOSS (BTU/HR) 10
0
1400
2800
4200
5600
7000
8400
9800
11200
12600
-0
4
8
12
16
20
24
28
TIME OF DAY
TOTAL
COND
INFIL
SOLAR
LIGHTS

Figure 3.1

(B)

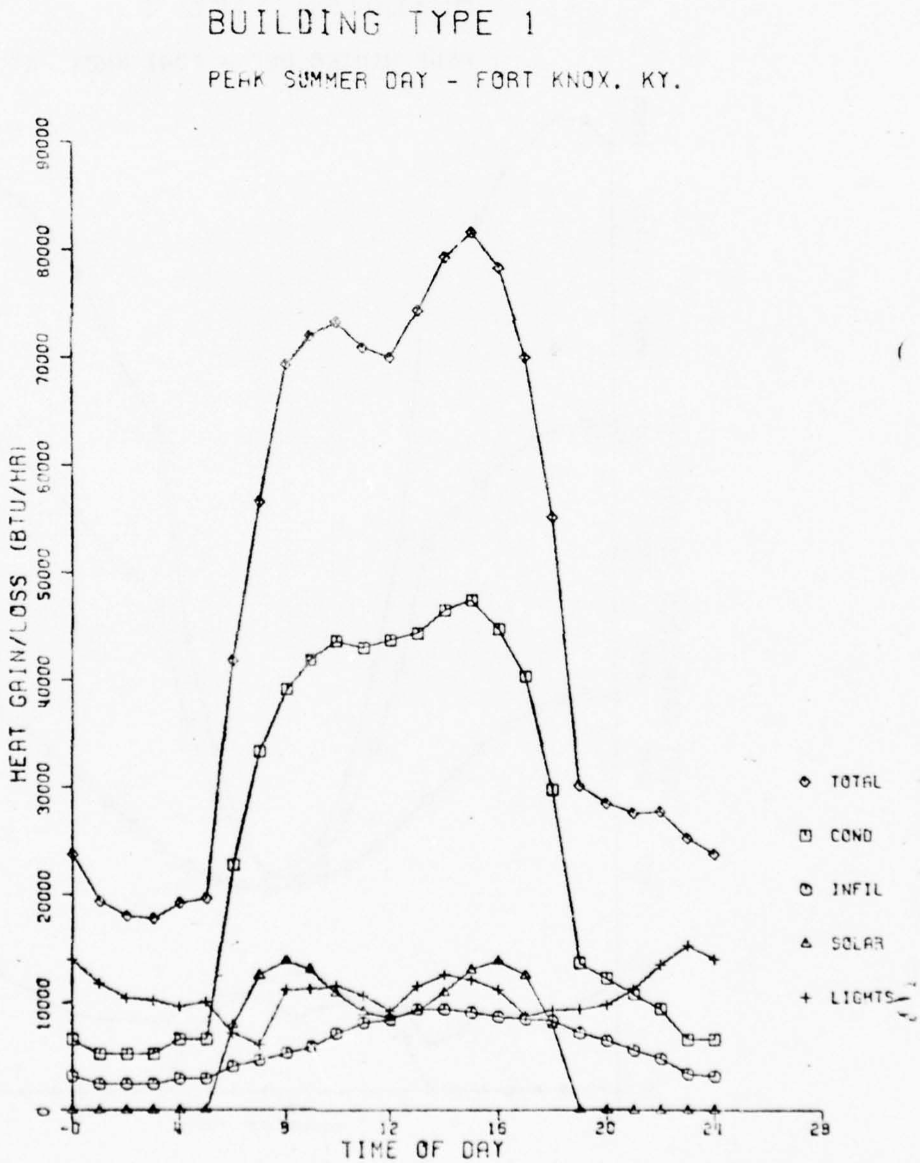

BUILDING TYPE 1
PEAK SUMMER DAY - FORT KNOX, KY.
HEAT GAIN/LOSS (BTU/HR)
0
10000
20000
30000
40000
50000
60000
70000
80000
90000
0
4
8
12
16
20
24
28
TIME OF DAY
TOTAL
COND
INFIL
SOLAR
LIGHTS

Figure 3.2

(A)

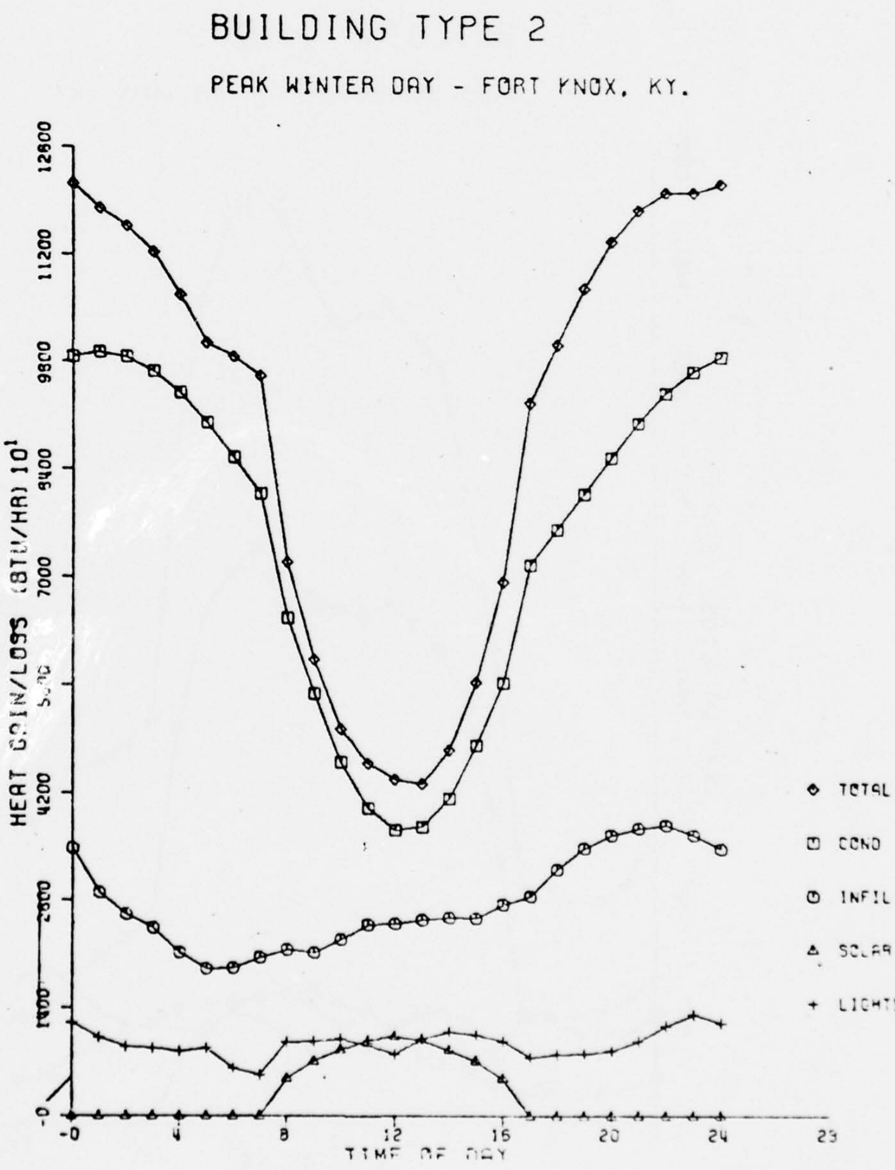

BUILDING TYPE 2
PEAK WINTER DAY - FORT KNOX, KY.
HEAT GAIN/LOSS (BTU/HR) 10¹
TIME OF DAY
TOTAL
COND
INFIL
SOLAR
LIGHTS
12600
11200
9800
8400
7000
4200
2800
1400
0
0
4
8
12
16
20
24
28

Figure 3.2

(B)

BUILDING TYPE 2

PEAK SUMMER DAY - FORT KNOX, KY.

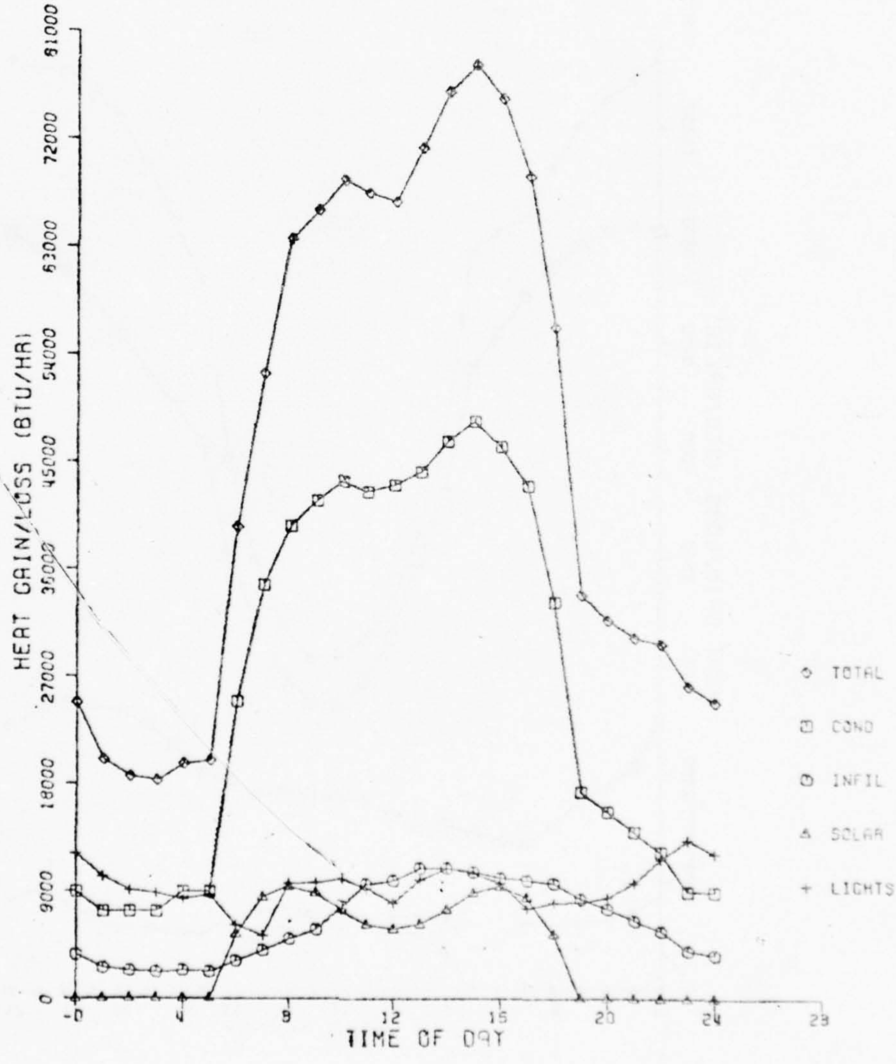


Figure 3.3

(A)

BUILDING TYPE 3

PEAK WINTER DAY - FORT KNOX, KY.

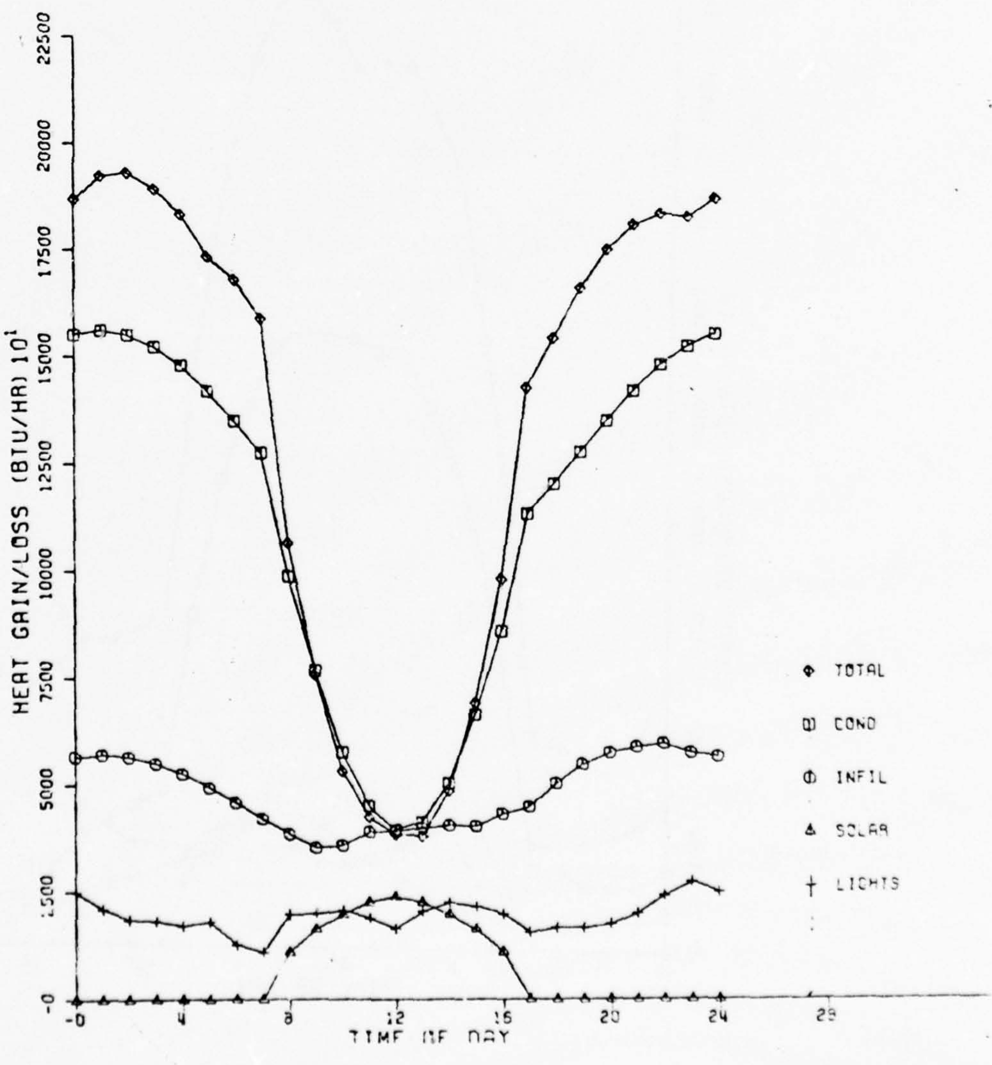


Figure 3.3

(B)

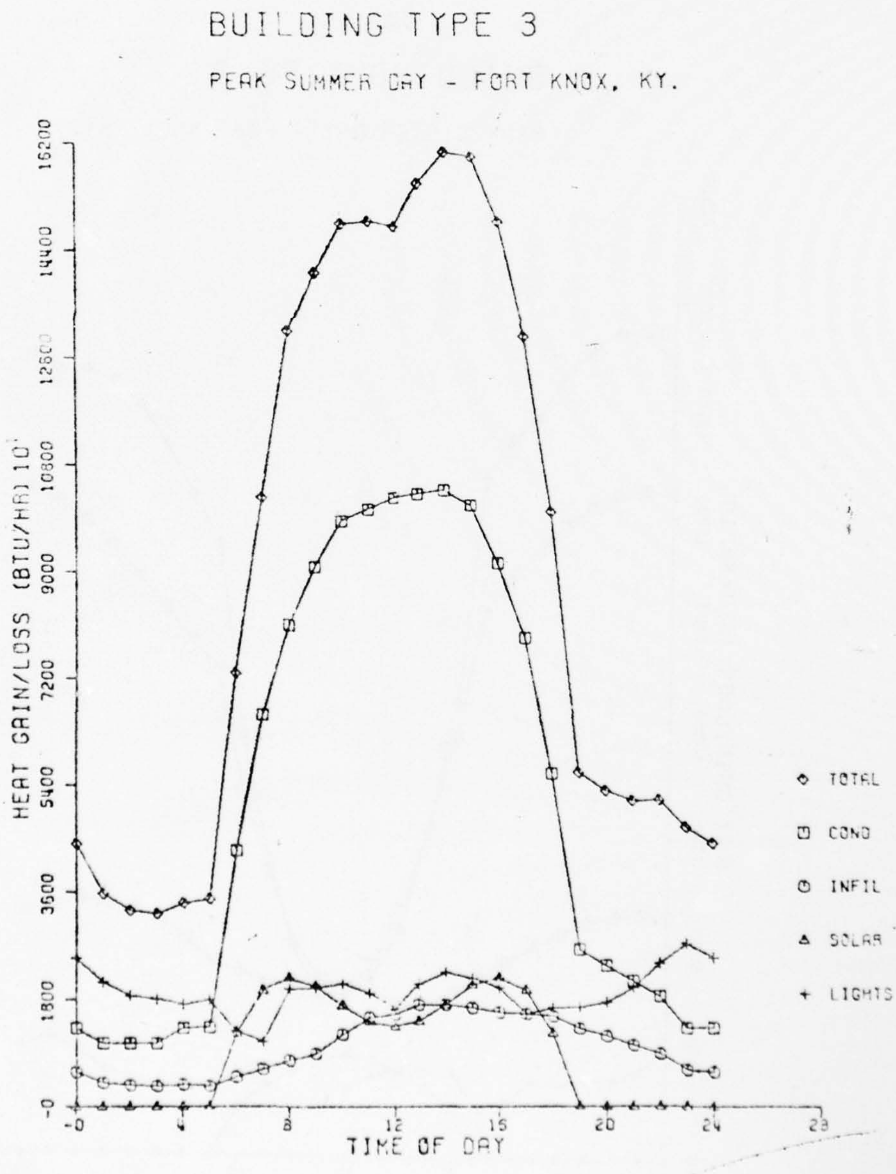

BUILDING TYPE 3
PEAK SUMMER DAY - FORT KNOX, KY.
HEAT GAIN/LOSS (BTU/HR) 10'
16200
14400
12600
10800
9000
7200
5400
3600
1800
0
TIME OF DAY
0
4
8
12
16
20
24
28
TOTAL
COND
INFIL
SOLAR
LIGHTS

Figure 3.4

(A)

BUILDING TYPE 4

PEAK WINTER DAY - FORT KNOX, KY.

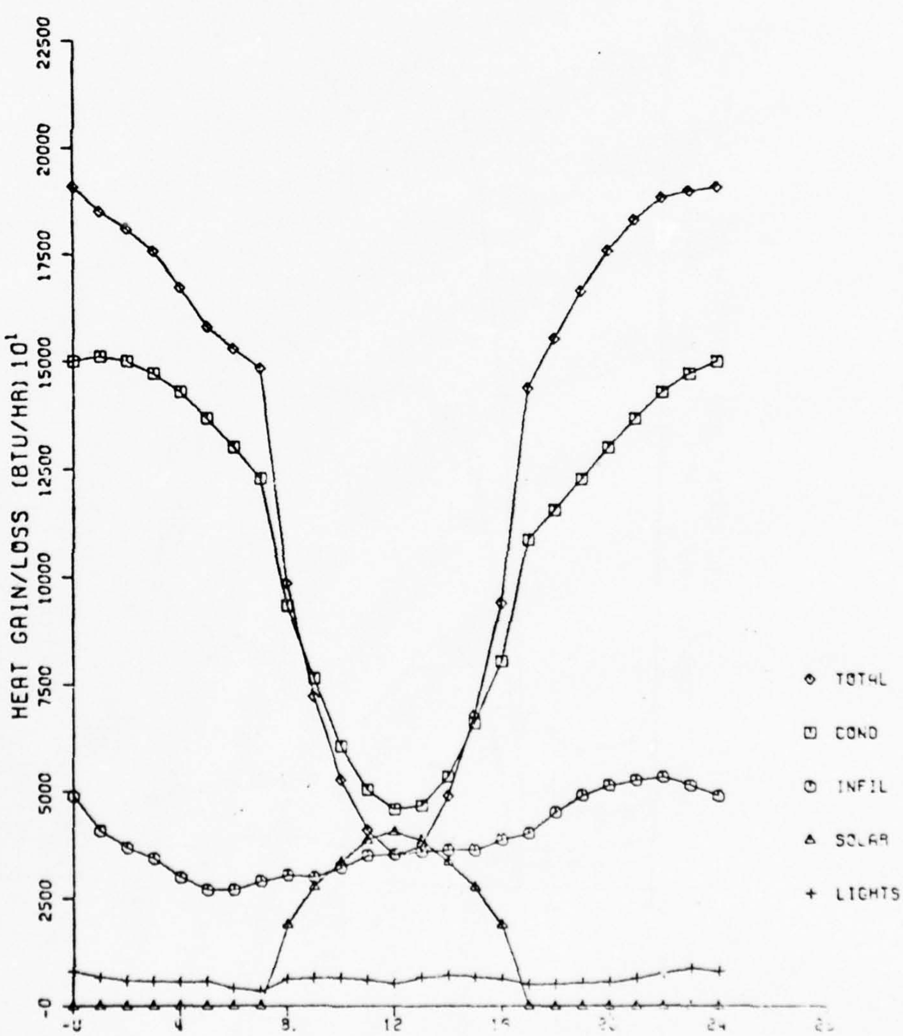


Figure 3.4

(B)

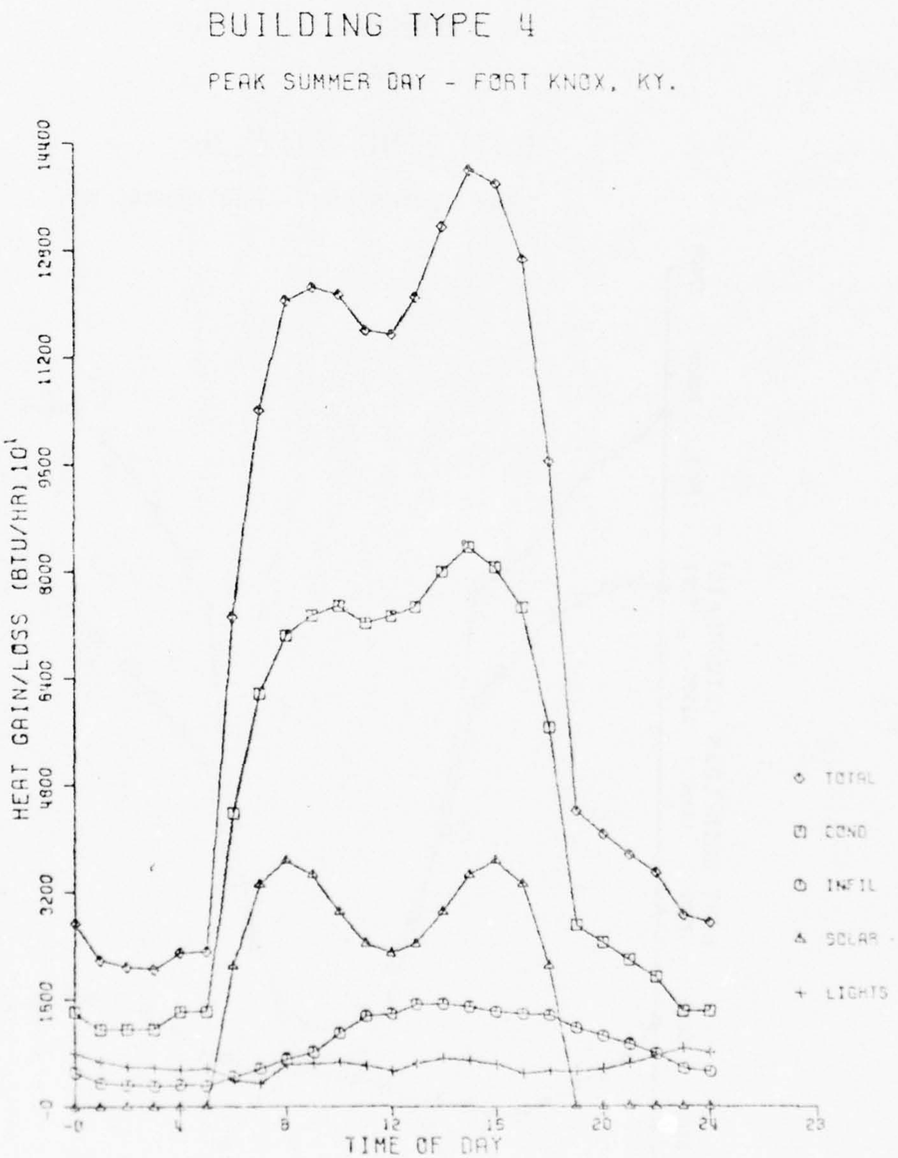

BUILDING TYPE 4
PEAK SUMMER DAY - FORT KNOX, KY.
HEAT GAIN/LOSS (BTU/HR) 10¹
14400
12900
11200
9600
8000
6400
4800
3200
1600
0
0
4
8
12
16
20
24
28
TIME OF DAY
TOTAL
COND
INFIL
SOLAR
LIGHTS

Figure 3.5

(A)

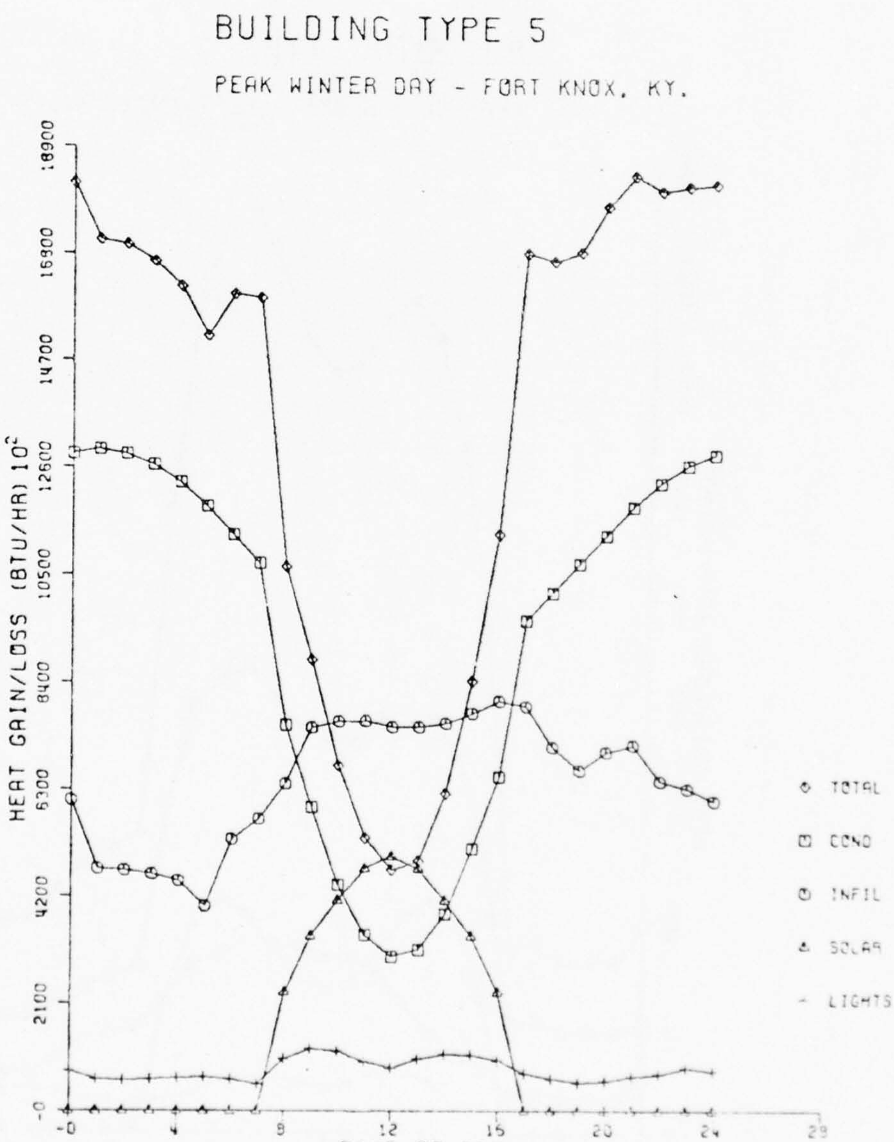

BUILDING TYPE 5
PEAK WINTER DAY - FORT KNOX, KY.
HEAT GAIN/LOSS (BTU/HR) 10^2
18900
16800
14700
12600
10500
8400
6300
4200
2100
0
-0
4
8
12
16
20
24
28
TOTAL
COND
INFIL
SOLAR
LIGHTS

Figure 3.5

(B)

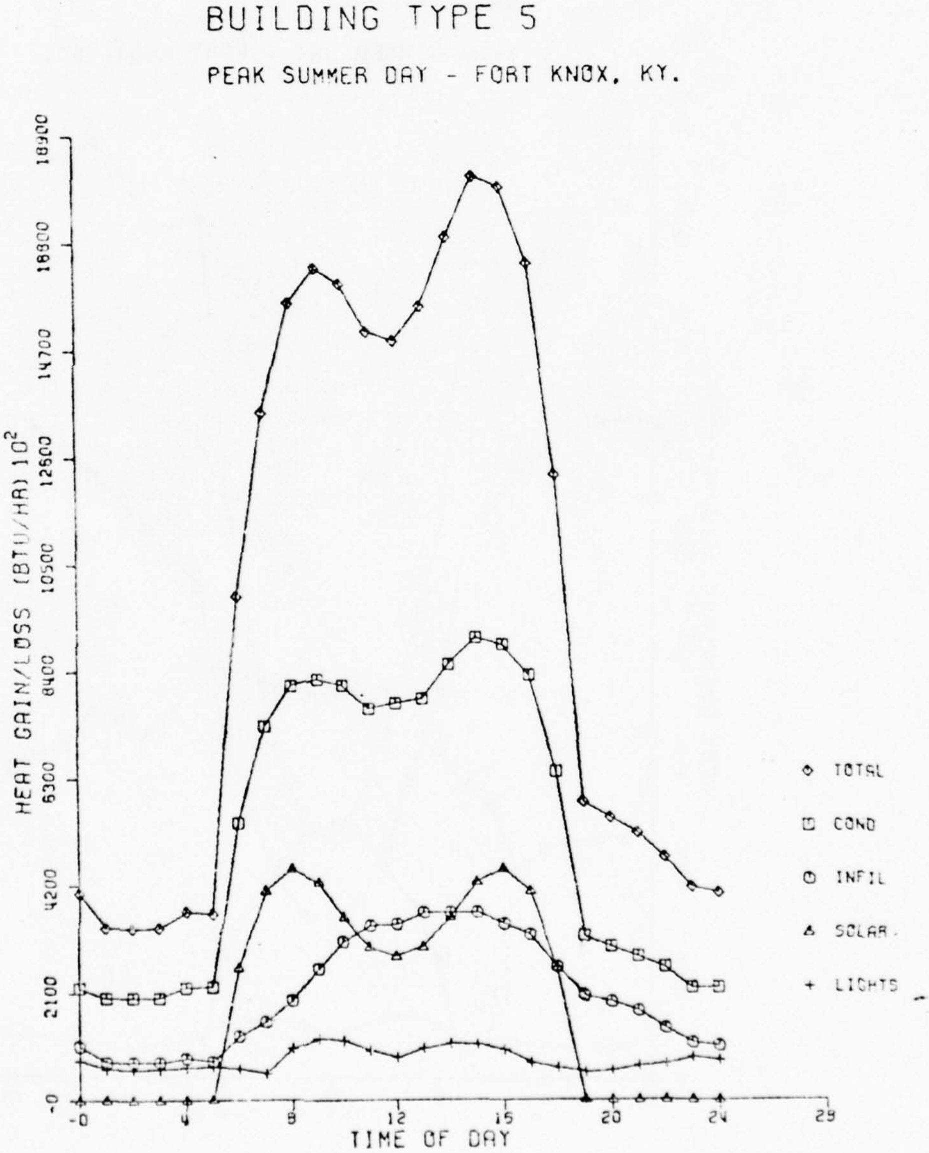

BUILDING TYPE 5
PEAK SUMMER DAY - FORT KNOX, KY.
HEAT GAIN/LOSS (BTU/HR) 10²
0
2100
4200
6300
8400
10500
12600
14700
16800
18900
-0
4
8
12
16
20
24
28
TIME OF DAY
TOTAL
COND
INFIL
SOLAR
LIGHTS

Figure 3.6

(A)

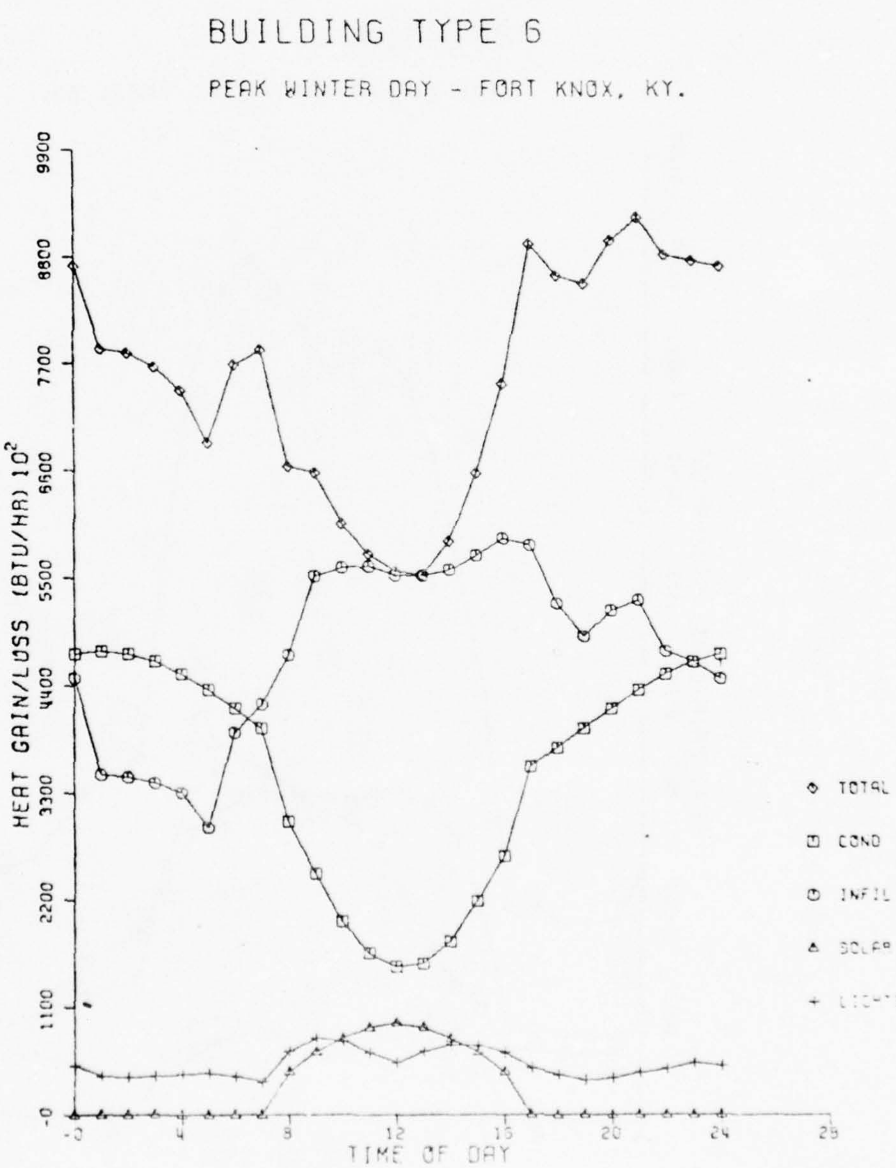

BUILDING TYPE 6
PEAK WINTER DAY - FORT KNOX, KY.
HEAT GAIN/LOSS (BTU/HR) 10^2
0
1100
2200
3300
4400
5500
6600
7700
8800
9900
TIME OF DAY
0
4
8
12
16
20
24
28
TOTAL
COND
INFIL

Figure 3.6

(B)

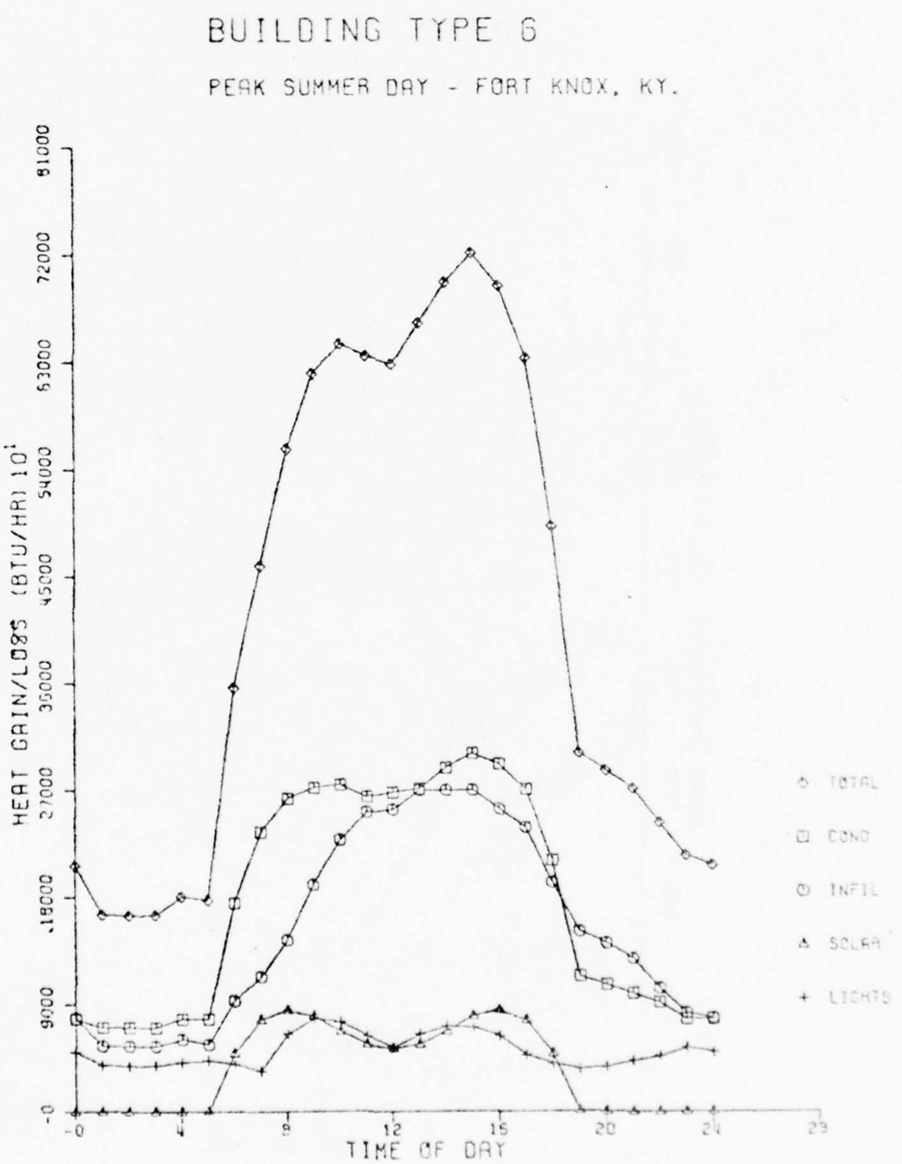

BUILDING TYPE 6
PEAK SUMMER DAY - FORT KNOX, KY.
HEAT GAIN/LOSS (BTU/HR) 10¹
TIME OF DAY
0
9000
18000
27000
36000
45000
54000
63000
72000
81000
0
4
8
12
16
20
24
28
TOTAL
COND
INFIL
SOLAR
LIGHTS

Figure 3.7

(A)

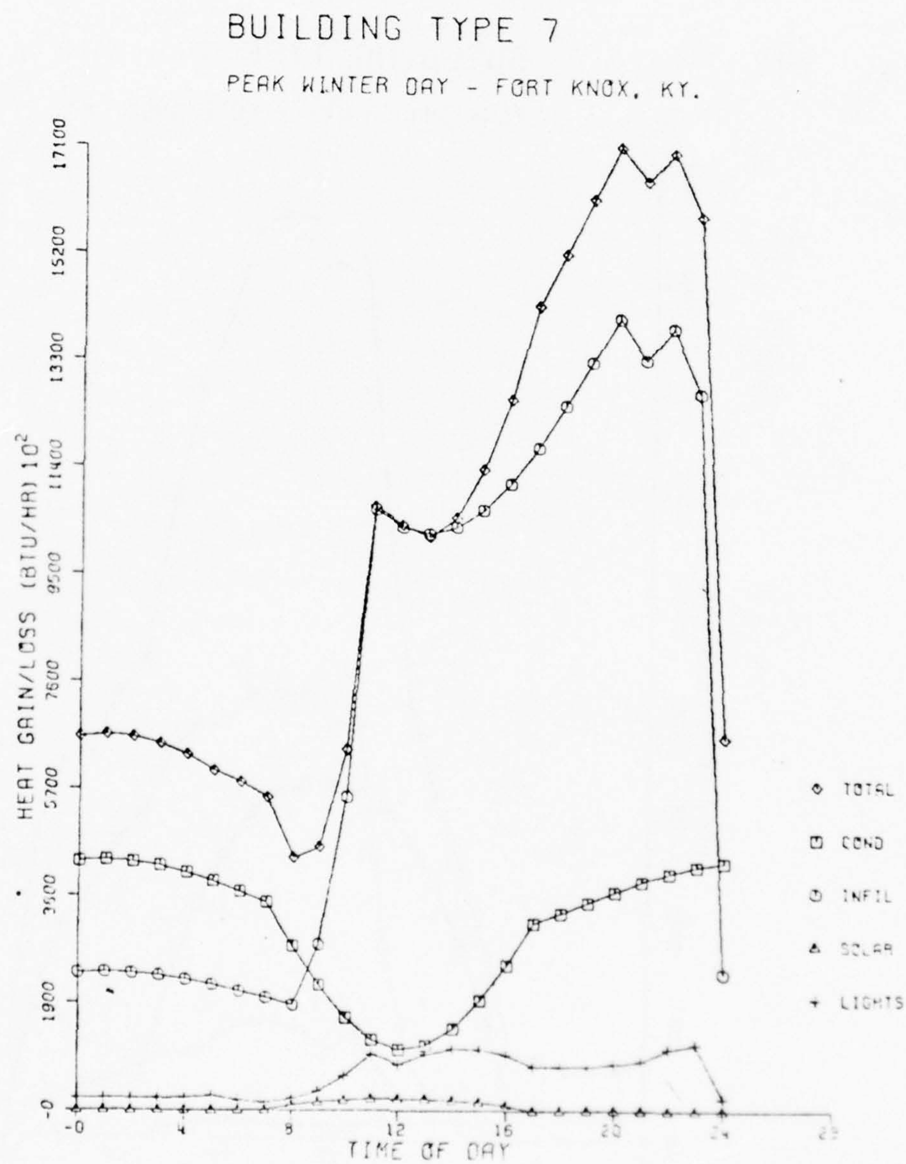
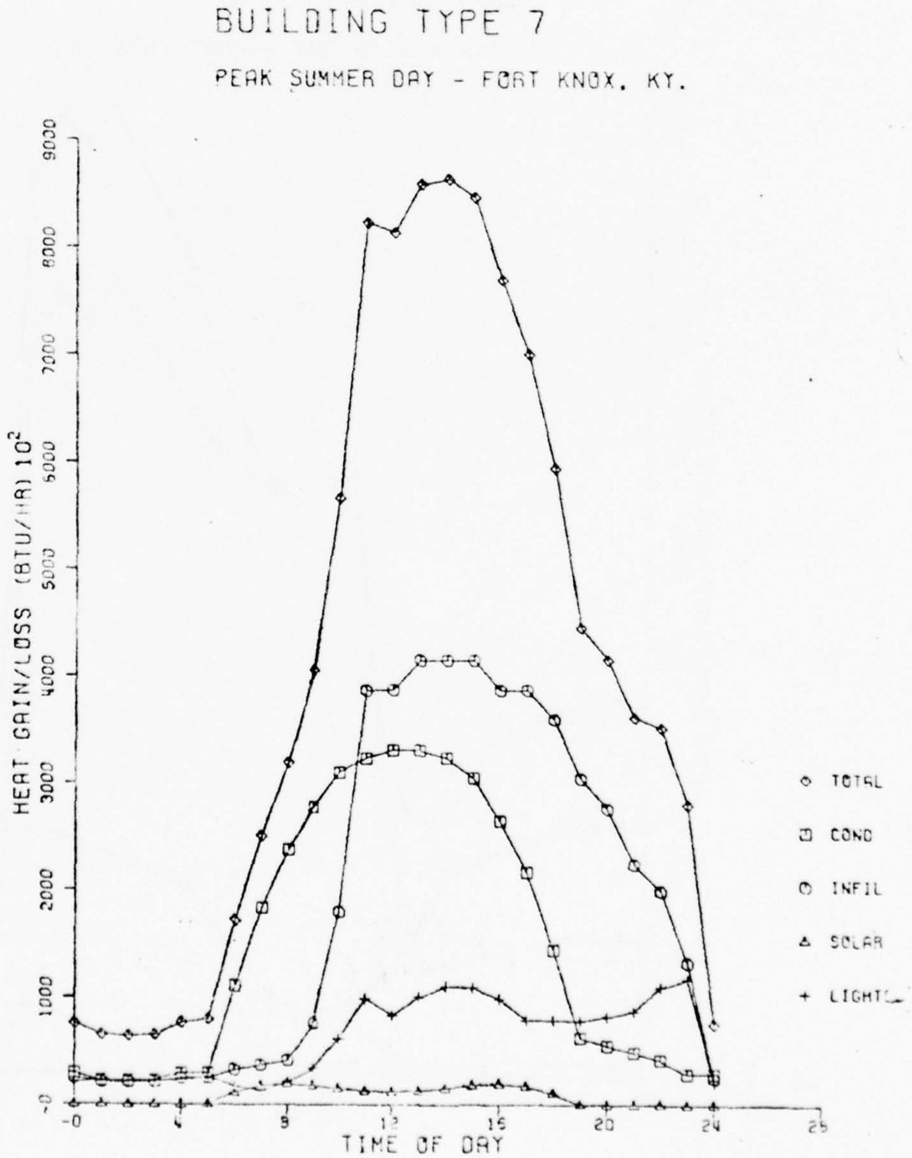

BUILDING TYPE 7
PEAK WINTER DAY - FORT KNOX, KY.
HEAT GAIN/LOSS (BTU/HR) 10^2
17100
15200
13300
11400
9500
7600
5700
3800
1900
0
-0
4
8
12
16
20
24
28
TIME OF DAY
TOTAL
COND
INFIL
SOLAR
LIGHTS

Figure 3.7

(B)



BUILDING TYPE 7
PEAK SUMMER DAY - FORT KNOX, KY.
HEAT GAIN/LOSS (BTU/HR) 10^2
TIME OF DAY
TOTAL
COND
INFIL
SOLAR
LIGHTS

Figure 3.8

(A)

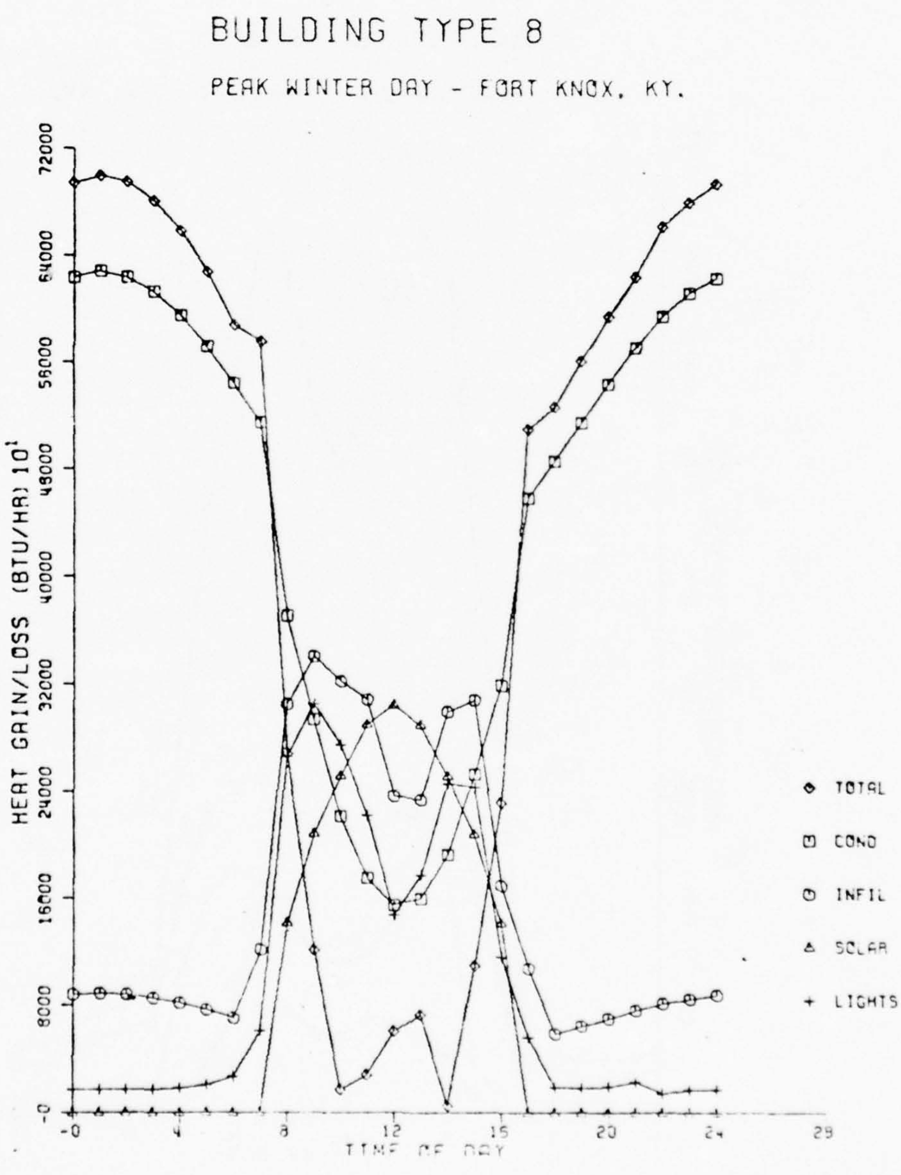

BUILDING TYPE 8
PEAK WINTER DAY - FORT KNOX, KY.
HEAT GAIN/LOSS (BTU/HR) 10¹
72000
64000
56000
48000
40000
32000
24000
16000
8000
-0
-0
4
8
12
16
20
24
28
TIME OF DAY
TOTAL
COND
INFIL
SOLAR
LIGHTS

Figure 3.8

(B)

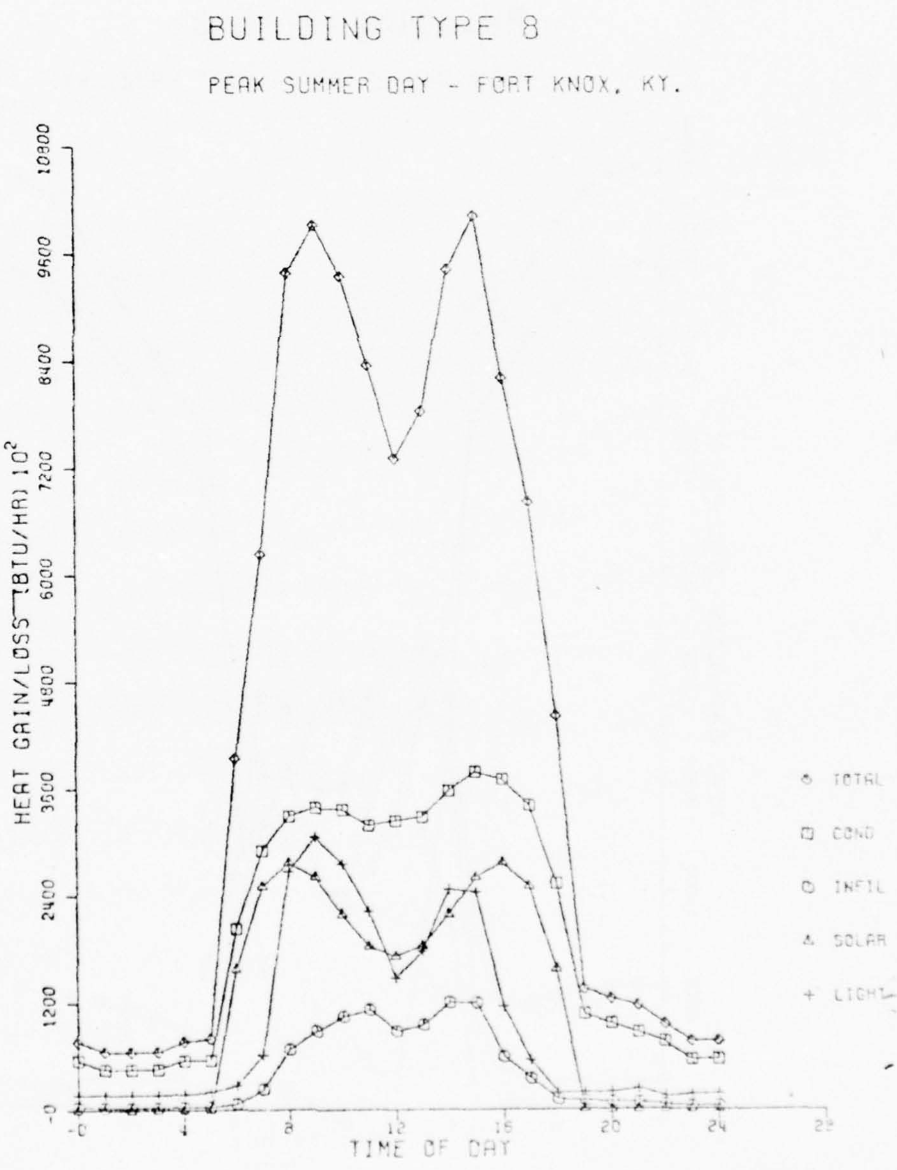

BUILDING TYPE 8
PEAK SUMMER DAY - FORT KNOX, KY.
HEAT GAIN/LOSS (BTU/HR) 10²
10800
9600
8400
7200
6000
4800
3600
2400
1200
TIME OF DAY
8
12
16
20
24
TOTAL
COND
INFIL
SOLAR
LIGHT

Figure 3.9

(A)

BUILDING TYPE 9

PEAK WINTER DAY - FORT KNOX, KY.

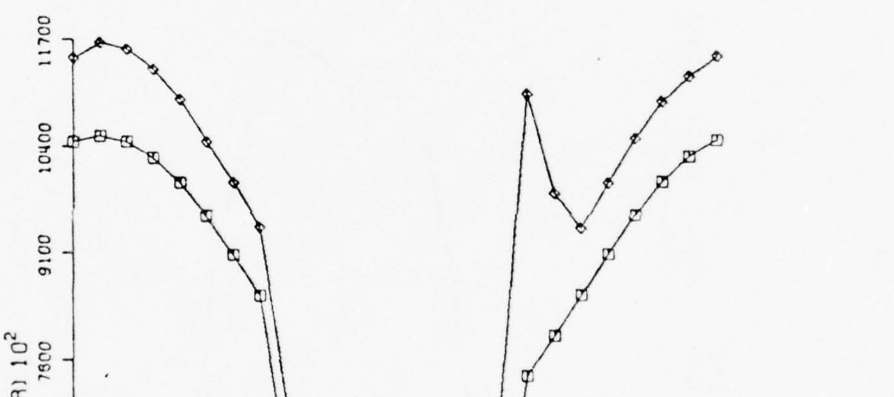
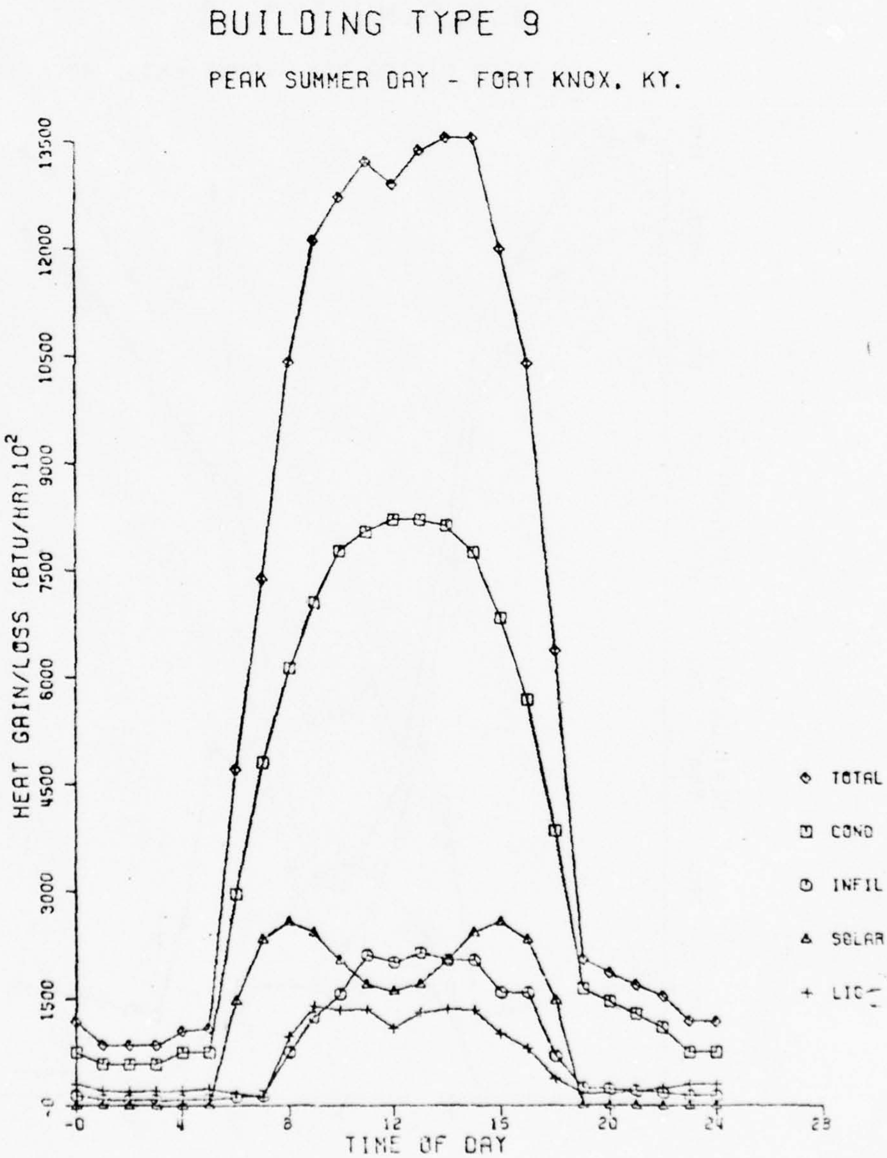


Figure 3.9

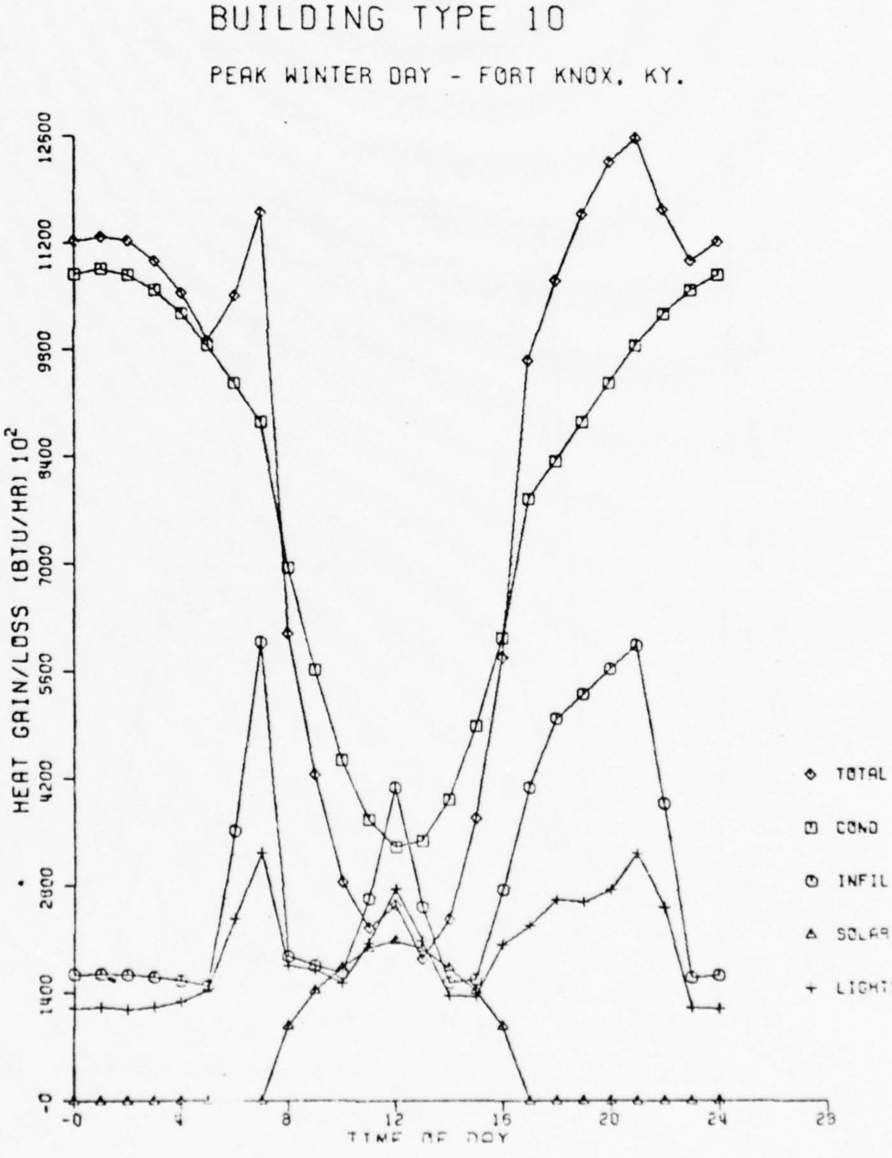
(B)



BUILDING TYPE 9
PEAK SUMMER DAY - FORT KNOX, KY.
HEAT GAIN/LOSS (BTU/HR) 10^2
13500
12000
10500
9000
7500
6000
4500
3000
1500
-0
-0
4
8
12
16
20
24
28
TIME OF DAY
TOTAL
COND
INFIL
SOLAR

Figure 3.10

(A)



BUILDING TYPE 10
PEAK WINTER DAY - FORT KNOX, KY.
HEAT GAIN/LOSS (BTU/HR) 10^2
12600
11200
9800
8400
7000
5600
4200
2800
1400
0
TIME OF DAY
0
4
8
12
16
20
24
28
TOTAL
COND
INFIL
SOLAR
LIGHTS

Figure 3.10

(B)

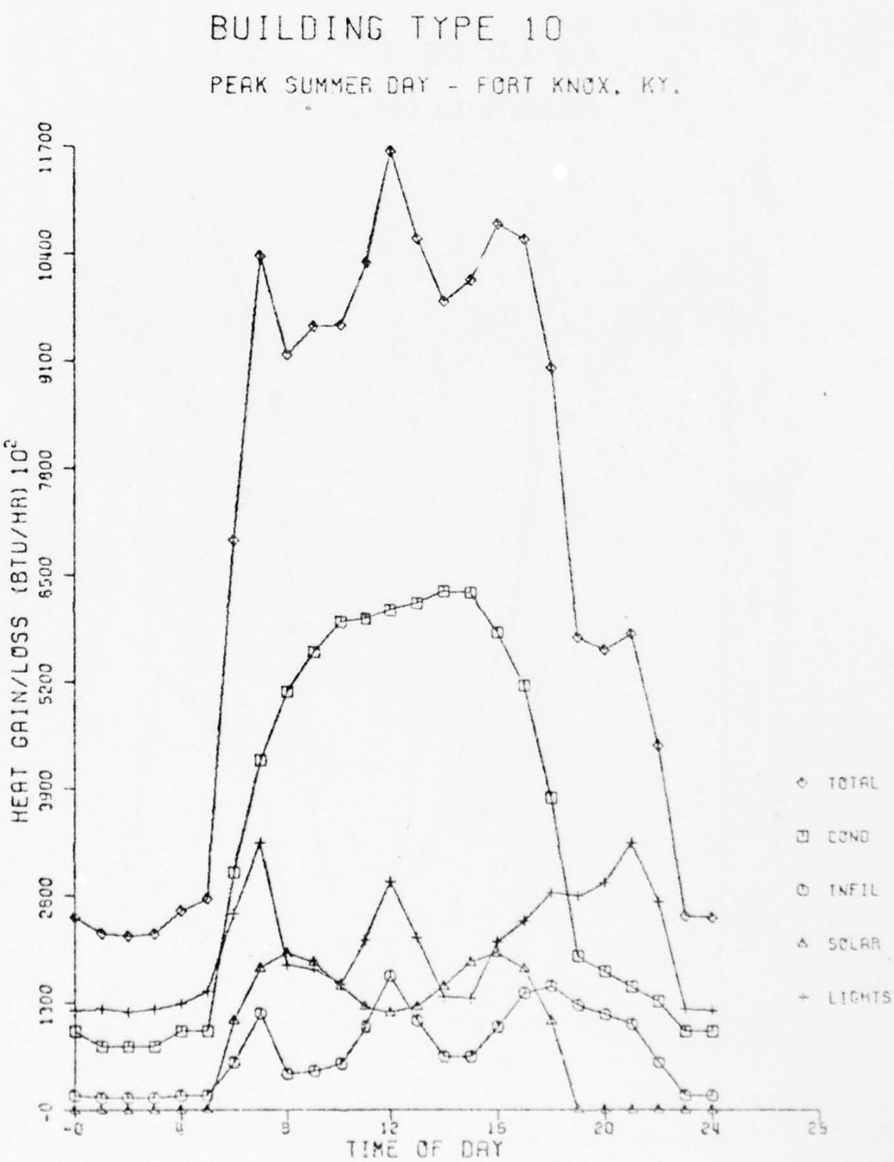

BUILDING TYPE 10
PEAK SUMMER DAY - FORT KNOX, KY.
HEAT GAIN/LOSS (BTU/HR) 10^2
11700
10400
9100
7800
6500
5200
3900
2600
1300
-0
-0
4
8
12
16
20
24
25
TIME OF DAY
TOTAL
COND
INFIL
SOLAR
LIGHTS

Figure 3.11

(A)

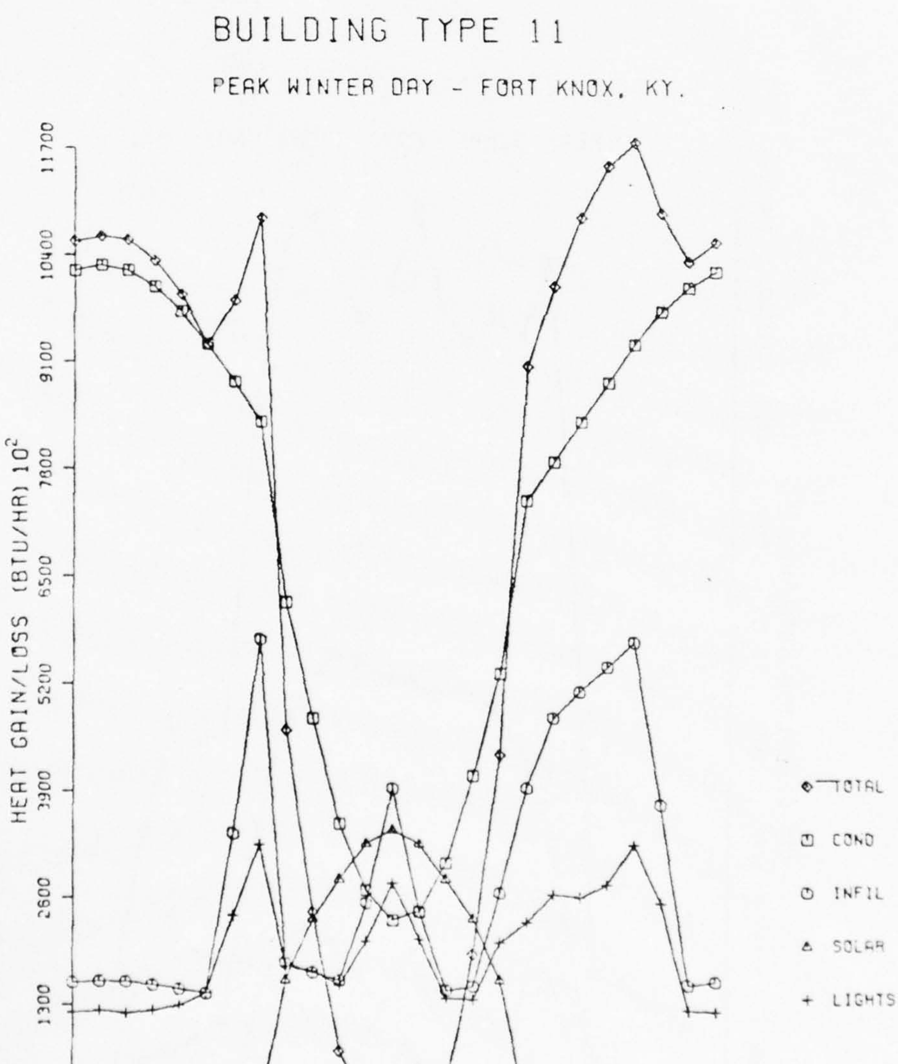

BUILDING TYPE 11
PEAK WINTER DAY - FORT KNOX, KY.
HEAT GAIN/LOSS (BTU/HR) 10^2
11700
10400
9100
7800
6500
5200
3900
2600
1300
-0
-0
4
8
12
16
20
24
28
TIME OF DAY
TOTAL
COND
INFIL
SOLAR
LIGHTS

Figure 3.11

(B)

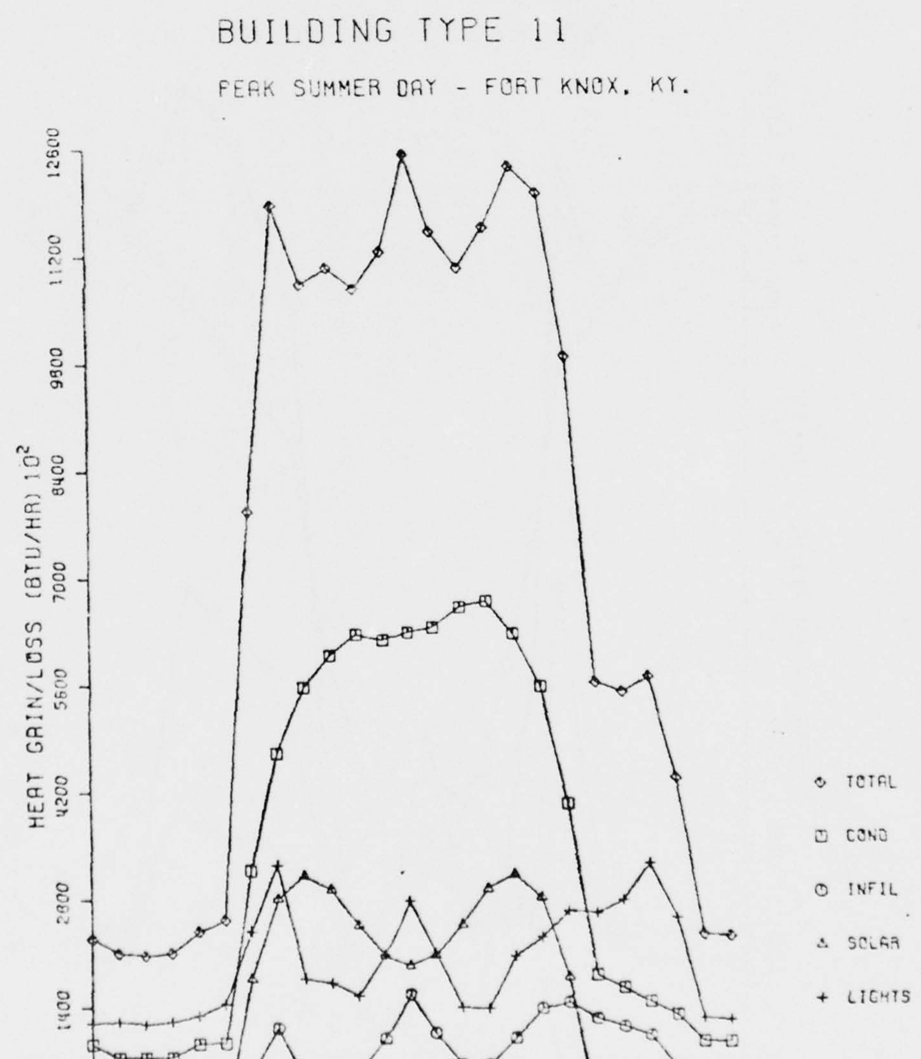

BUILDING TYPE 11
PEAK SUMMER DAY - FORT KNOX, KY.
HEAT GAIN/LOSS (BTU/HR) 10^2
0
1400
2800
4200
5600
7000
8400
9800
11200
12600
0
4
8
12
16
20
24
28
TOTAL
COND
INFIL
SOLAR
LIGHTS

Figure 3.12

(A)

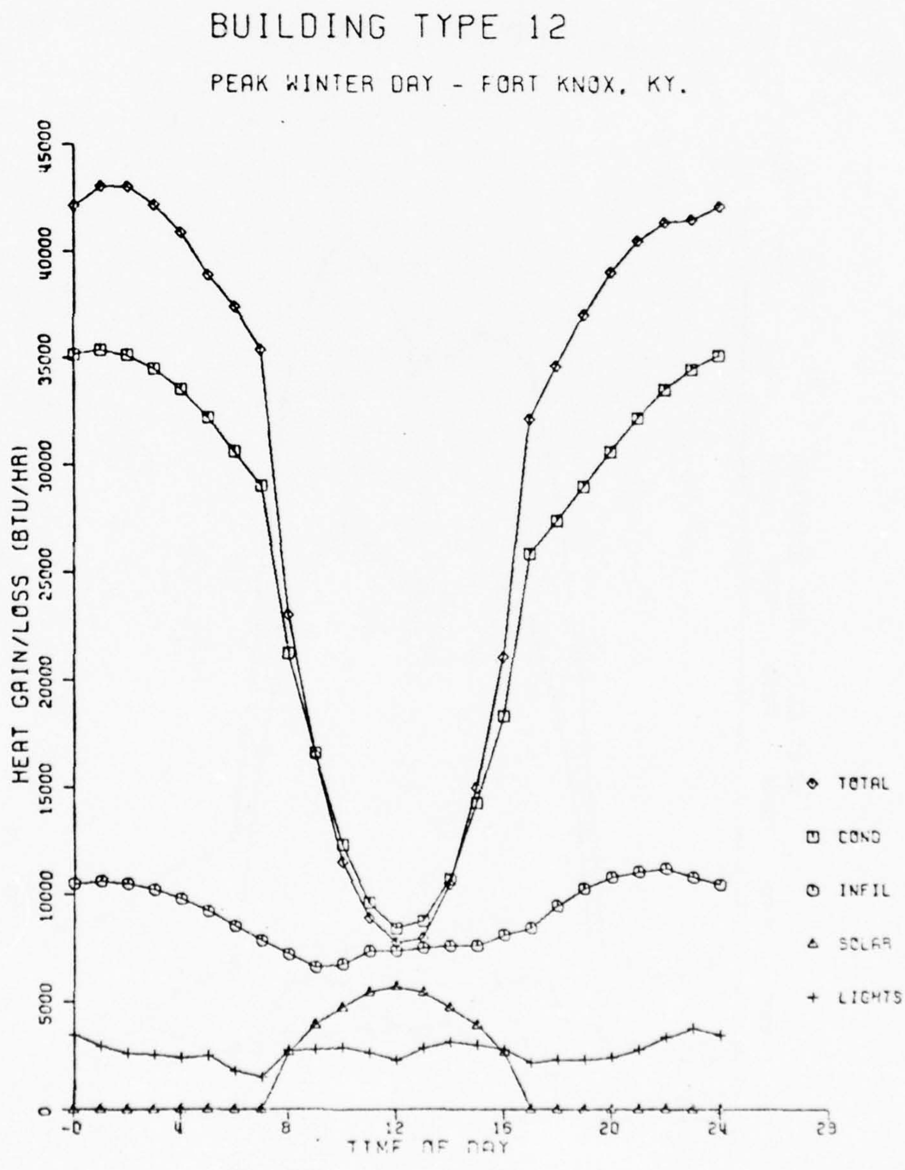

BUILDING TYPE 12
PEAK WINTER DAY - FORT KNOX, KY.
HEAT GAIN/LOSS (BTU/HR)
0
5000
10000
15000
20000
25000
30000
35000
40000
45000
-0
4
8
12
16
20
24
28
TIME OF DAY
TOTAL
COND
INFIL
SOLAR
LIGHTS

Figure 3.12

(B)

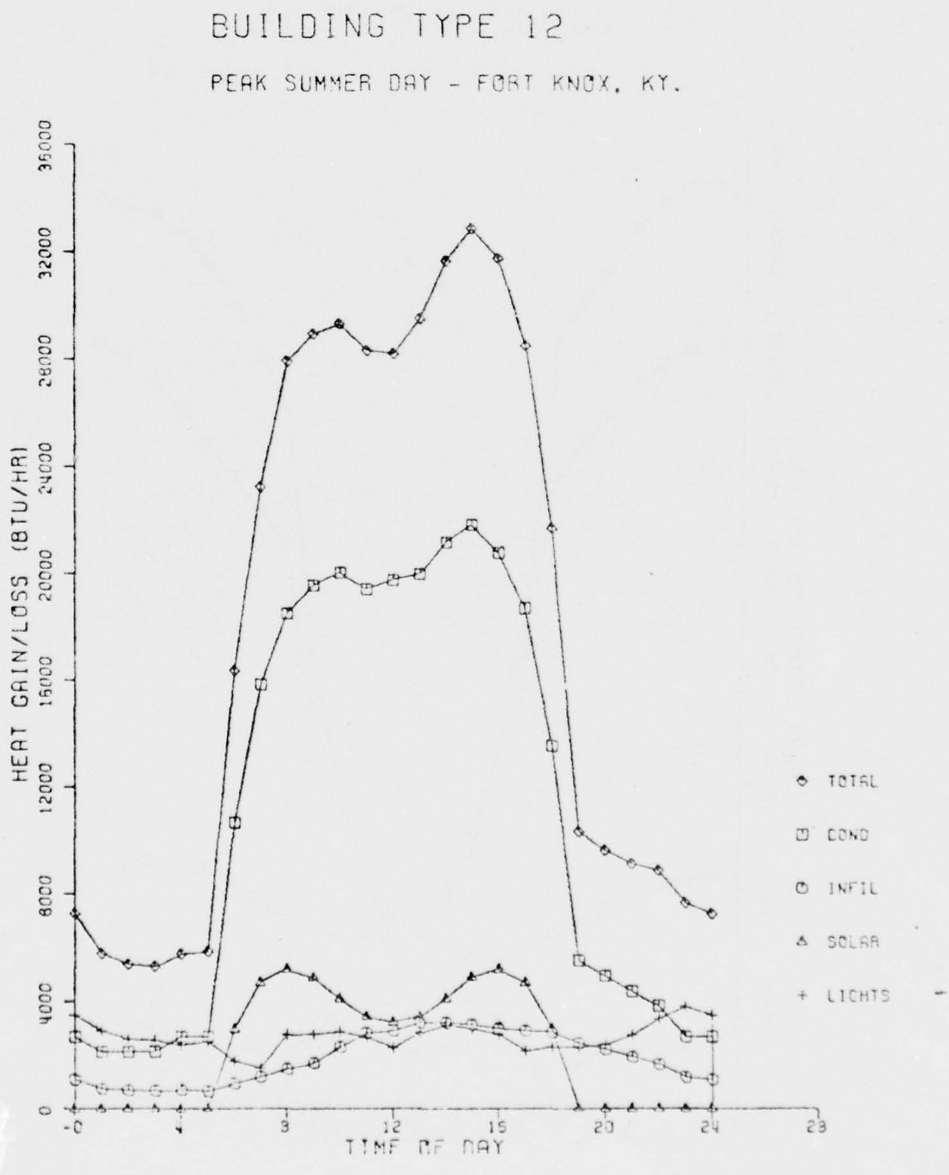

BUILDING TYPE 12
PEAK SUMMER DAY - FORT KNOX, KY.
HEAT GAIN/LOSS (BTU/HR)
TIME OF DAY
TOTAL
COND
INFIL
SOLAR
LIGHTS

Figure 3.13

(A)

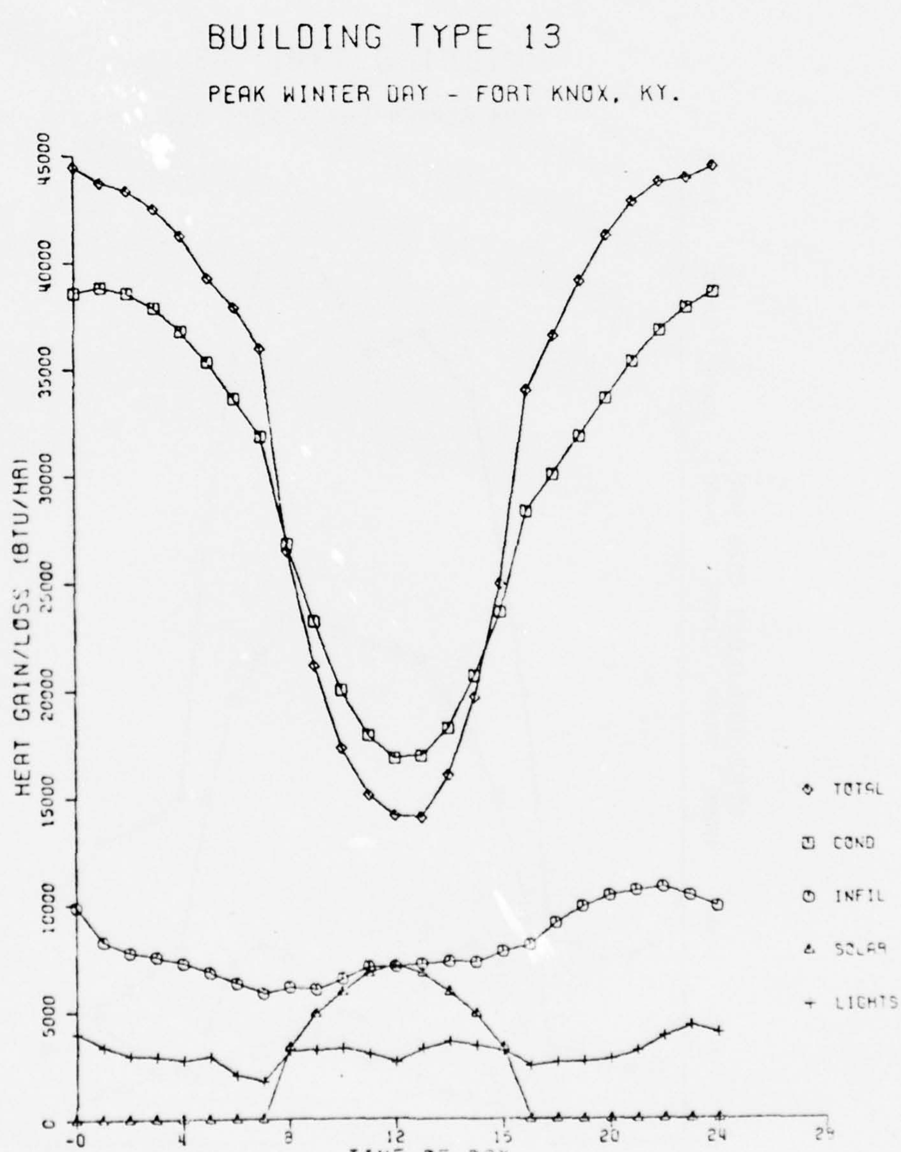

BUILDING TYPE 13
PEAK WINTER DAY - FORT KNOX, KY.
HEAT GAIN/LOSS (BTU/HR)
45000
40000
35000
30000
25000
20000
15000
10000
5000
0
-0
4
8
12
16
20
24
TOTAL
COND
INFIL
SOLAR
LIGHTS

Figure 3.13

(B)

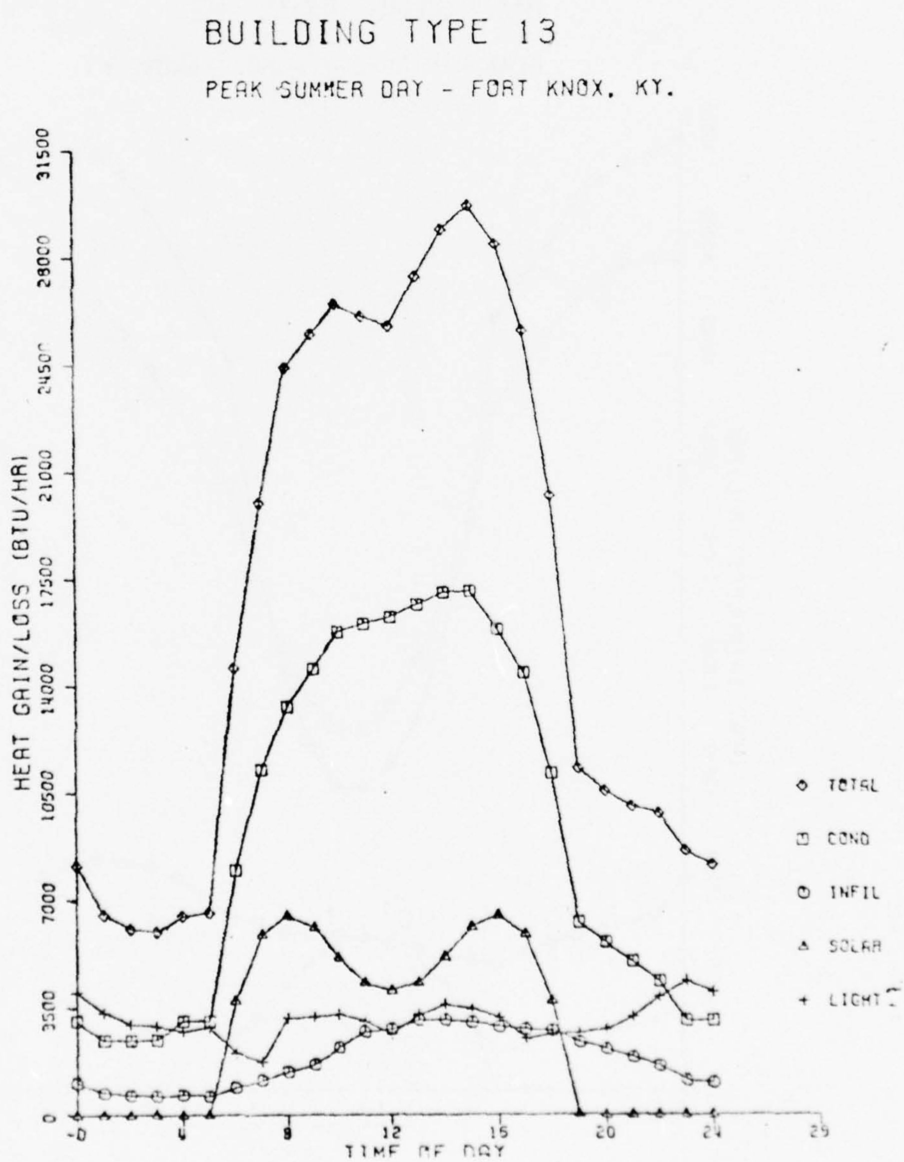

BUILDING TYPE 13
PEAK SUMMER DAY - FORT KNOX, KY.
HEAT GAIN/LOSS (BTU/HR)
0
3500
7000
10500
14000
17500
21000
24500
28000
31500
0
4
8
12
16
20
24
28
TIME OF DAY
TOTAL
COND
INFIL
SOLAR
LIGHT

Figure 3.14

(A)

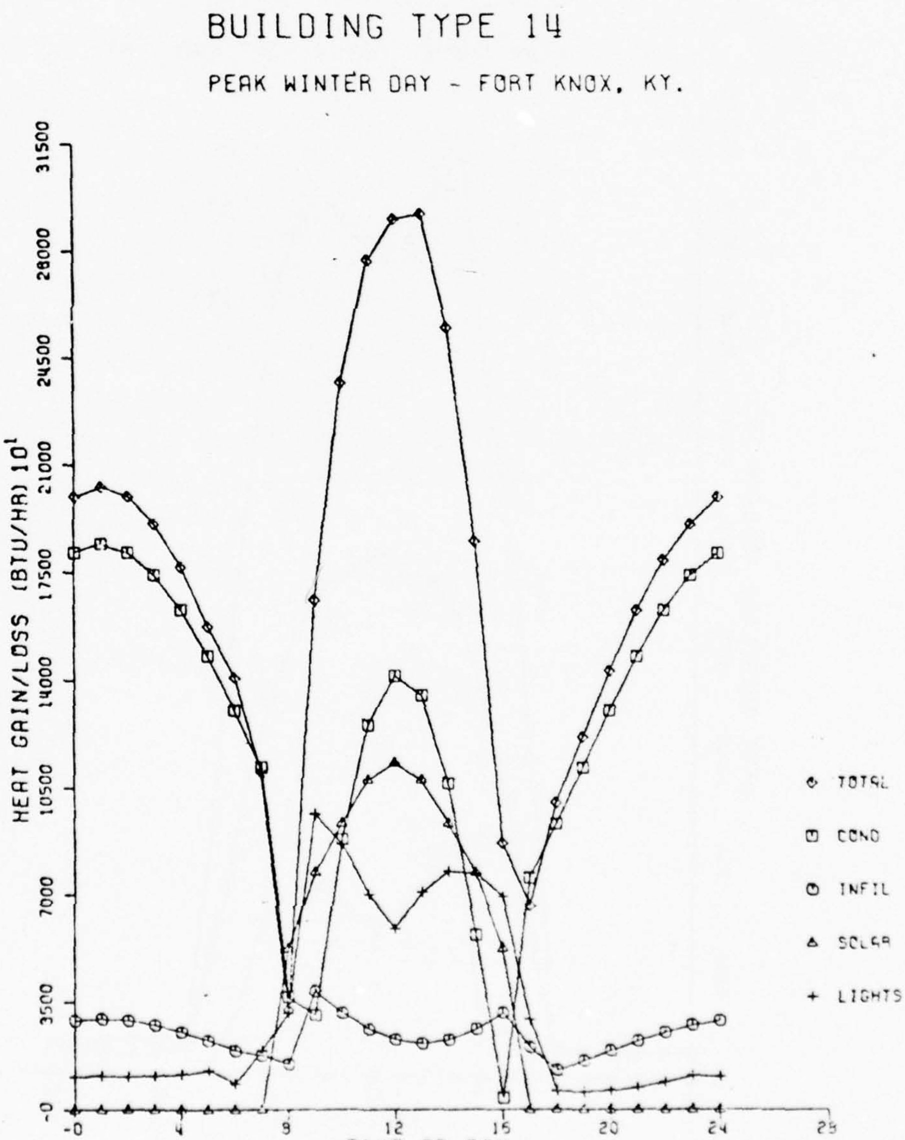

BUILDING TYPE 14
PEAK WINTER DAY - FORT KNOX, KY.
HEAT GAIN/LOSS (BTU/HR) 10^1
31500
28000
24500
21000
17500
14000
10500
7000
3500
0
0
4
8
12
16
20
24
28
TOTAL
COND
INFIL
SOLAR
LIGHTS

Figure 3.14

(B)

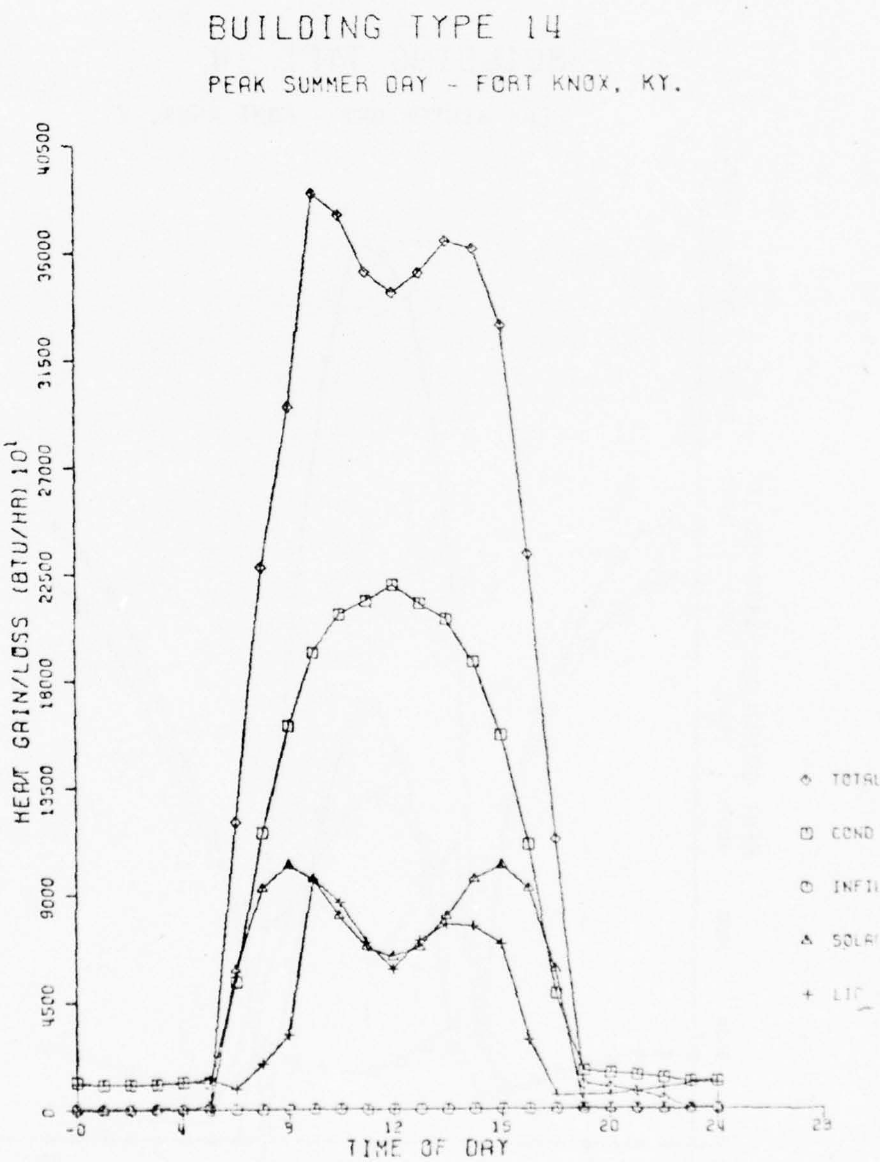

BUILDING TYPE 14
PEAK SUMMER DAY - FORT KNOX, KY.
HEAT GAIN/LOSS (BTU/HR) 10[1]
TIME OF DAY
0
4500
9000
13500
18000
22500
27000
31500
36000
40500
-0
4
9
12
15
20
24
TOTAL
COND
INFIL
SOLAR

Figure 3.15

(A)

BUILDING TYPE 15

PEAK WINTER DAY - FORT KNOX, KY.

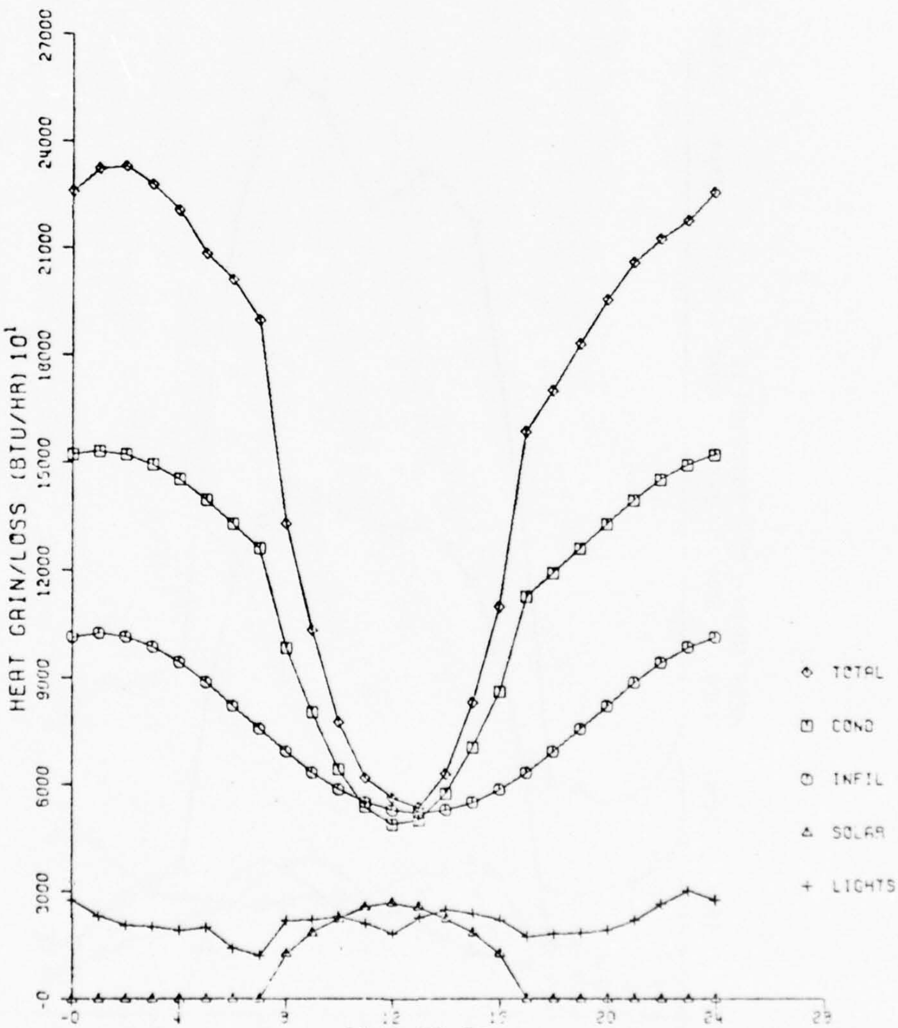
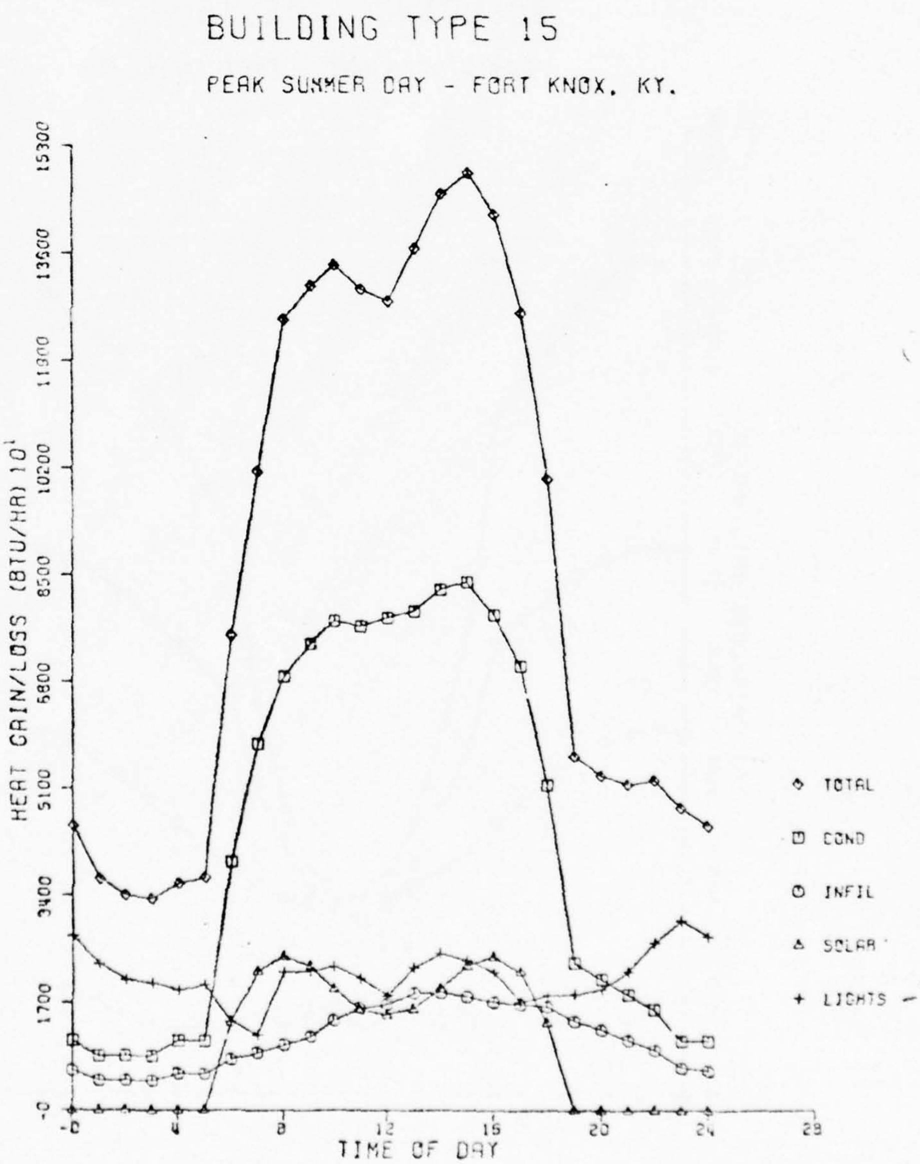


Figure 3.15

(B)


BUILDING TYPE 15
PEAK SUMMER DAY - FORT KNOX, KY.
HEAT GAIN/LOSS (BTU/HR) 10¹
15300
13600
11900
10200
8500
6800
5100
3400
1700
-0
-0
4
8
12
16
20
24
28
TIME OF DAY
TOTAL
COND
INFIL
SOLAR
LIGHTS

day reduce that building's heating load to zero, even though the outside air temperature is only 32° to 33 °F. Consultation with Army personnel at FESA and an independent analysis by Michael Baker, Jr. of New York, Inc. [3] have verified this behavior for these particular buildings.) All forced air ventilation and induced infiltration air flows are assumed to be direct air exchanges between the interior and exterior of the buildings. The significant effects of these components of the space conditioning loads are evident in the winter demand schedules for the hospitals (Figs. 3.5-3.6), the ventilation requirements of which are large and the usage of which is fairly constant throughout the day, and for the community buildings (Fig. 3.7), the afternoon and evening usage and large ventilation requirements of which during occupancy cause both its summer and winter demand curves to be skewed slightly more toward the evening hours than those of the other building types. (The winter day profile for the hospitals, while irregular, is relatively flat compared with those of the other categories due to the hospital's fairly uniform occupancy characteristics and the offsetting effects of slightly higher ventilation requirements and solar heating during the day; the large variation in its summer day demand occurs due to the additive effects of these components when the ambient air is at a higher temperature than that desired within the building). The Storage (Fig. 3.14) and

Operations (Fig. 3.9) building categories are assumed to have no air conditioning (see Appendix B). Since this condition is transmitted to the TDIST2 consumer demand models only by requiring the internal room temperatures to vary directly with the outside air temperature, during periods of sunlight the combined effects of solar heating and a small amount of internal lighting produce the nominal summer day cooling demands shown. As has been mentioned previously, these demands are applied to the energy supply systems only through the use of specified space conditioning equipment units. Since no air conditioning is desired for these two categories, their cooling loads do not appear on the system. The program is arranged not to provide cooling during the heating season and heating during the cooling season. The demand profiles for all the Troop Housing and Family Housing categories exhibit the same qualitative behavior, reflecting the general similarity of occupancy of these units during the late afternoon and evening hours and the dominance of solar heating during the day.

REFERENCES

1. Goldman, S.G., Best, F.R., Golay, M.W. "TDIST2, A Computer Program for Community Energy Consumption Analysis and Total Energy System Design," Project Report, Contract No. DAAK02-74-C-0308, Department of Nuclear Engineering, MIT, June 1977.

2. Kusuda, T., "SUN, An Algorithm to Find Solar Position, and Intensity of Direct Normal and Diffuse Radiation," NBSLD, Computer Program for Heating and Cooling Loads in Buildings, NBSIR-74-574, National Bureau of Standards, Washington, D.C., (Nov. 1974), pp. 6a-14a.

3. "Administration Building," Section 10, Energy Conservation Study - Fort Bragg, North Carolina, Department of the Army Contract, DACA73-74-C-0011, Michael Baker, Jr. of New York, Inc. Consulting Engineers, (April 1974).

CHAPTER 4

FORT KNOX THERMAL UTILITY SYSTEM OPTIONS

Encompassing an area of approximately 9 square miles, the inhabited section of Fort Knox occupies the southwestern end of the base. (The remainder of the base is used for training grounds, firing ranges, etc., and has very few permanent buildings.) The population of Fort Knox in 1985 is expected to be approximately 18,000. Supplying the residents with both thermal and electrical energy from a single power plant requires carefully designed piping systems and electrical distribution circuits to deliver the necessary energy at a minimum total cost.

Figure 4.1 is a planning map for Fort Knox illustrating the layout of the inhabited area of the base as it is expected to appear in 1985. The crosshatched areas are World War II vintage temporary buildings which are scheduled to be removed by 1985.

One of the ground rules established early in the Fort Knox study was that the proposed TES be nominally capable of supplying the base's total annual energy demands without relying upon any auxiliary capacity from outside the Fort's boundaries. Because some loads (space conditioning, domestic hot water) readily lend themselves to either thermal or electrical energy supplies, the possibility arises for

Figure 4.1

PLAN MAP FORT KNOX

☐ - indicates primary-to-secondary heat exchanger

◯ - indicates load center

optimizing the TES to obtain the highest average efficiency and lowest total cost through carefully designed tradeoffs between the percentages of the consumers utilizing the power plant's thermal and electrical energy outputs. Ideally, the optimal system design would be that configuration which, throughout the entire year, would cause electrical and thermal energy to be produced and consumed in a ratio such that none of the power plant's total energy output would be wasted. (Chapter 6 discusses more fully this optimization problem and its practical design limitations.) As demand variables for the TES design process, three general load categories are specified: space conditioning served thermally or electrically, domestic hot water service supplied thermally or electrically, and non-space-conditioning electrical demands (motors, appliances, lighting, etc.). The space conditioning and domestic hot water demands are computed on an individual building unit basis to allow their supply modes and service equipment to be varied. The non-space-conditioning electrical demand is taken from actual metering equipment at Fort Knox. Meter readings from May are used, since space conditioning equipment use is then at a minimum and therefore readings would truly represent non-space conditioning electrical demand. The values used for the base are shown in Table 4.1.

The minimum cost over life TES for Fort Knox is found by use of TDIST2 [1], the improved energy simulation

Table 4.1

FORT KNOX, NON-SPACE CONDITIONING ELECTRICAL DEMAND

Time	MW(e)
12	9.0
1	8.1
2	7.5
3	7.5
4	7.5
5	8.1
6	9.0
7	11.1
8	13.7
9	15.6
10	15.0
11	15.3
12	14.7
13	15.0
14	15.0
15	14.7
16	13.6
17	11.4
18	11.1
19	10.8
20	11.4
21	13.3
22	12.6
23	10.5
24	9.0

computer program which is the successor to TDIST [2], used in the earlier Fort Bragg study [3]. Using the building descriptions and weather data presented in Chapter 3, TDIST2 applies the calculated consumer energy demands to electrical and hot water distribution networks. The space conditioning demand of a building may be supplied by Thermal Utility System (TUS) hot water, by absorptive air conditioning or chilled water, by heat pumps, by compressive air conditioning or by electrical resistance heaters. Domestic hot water (sinks, washers, etc.) is supplied either by TUS hot water heating or by electrical resistance heaters.

The programmer may run TDIST2, using various proportions of consumer end use equipment, to tailor the load supplied by the power plant of the TES. For example, a group of buildings may require a certain amount of heating. This heating may be supplied by TUS hot water, by heat pumps or by electrical resistance heaters. Using resistance heaters will require a certain amount of electricity, using heat pumps will require less electricity (assuming coefficient of performance (COP) values >1) and using TUS hot water may involve no additional energy consumption if central station waste heat is available. Of course, each of these energy supply options will have different costs. TDIST2 is used to determine the mix of energy supply options which results in a minimum present-worth cost Total Energy System configuration. The cost of the TES includes plant capital

costs, fuel costs for a thirty year plant life, the Thermal Utility System capital costs, base electrical transmission and distribution costs, as well as end use equipment costs (heat pumps, heat exchangers, etc.). A complete description of the TDIST2 code is found in the report TDIST2: User's Manual [1].

The TES options considered for the analysis of Fort Knox are:

Option 1. Heating space conditioning is supplied by TUS hot water or by heat pumps. No electrical resistance heaters are used for space conditioning. Cooling is supplied either by chilled water supplied by the TUS; for buildings not on the TUS it is supplied by heat pumps running in the cooling mode. Domestic hot water is supplied everywhere by electrical resistance heaters.

Option 2. Heating space conditioning is supplied by TUS hot water or by heat pumps. No electrical resistance heaters are used for space conditioning. Cooling is supplied either by compressive air conditioners or by heat pumps running in the cooling mode. Domestic hot water is heated by TUS hot water; or for buildings not on the TUS by electrical resistance water heaters.

The percentage of the total base's winter design day space conditioning load supplied by the TUS is described as the TUS's Thermal/Electrical load split value. The Thermal/Electrical load split value refers to the percentage

of the peak winter design load the TUS is supplying. Hence, a Thermal/Electrical load split value of 100% means that all buildings have their space conditioning supplied completely by the TUS. A Thermal/Electrical load split value of 0% means that all space conditioning is being supplied electrically. Intermediate values of Thermal/Electrical load split refer to various mixes of thermally and electrically supplied space conditioning.

The 100% Thermal/Electrical split value for either TUS option requires that each building on Ft. Knox be attached by piping to the TUS. Figure 4.1 shows the TUS layout for the 100% split value. The dashed lines indicate 380°F hot water transmission piping (Primary Loop). The dotted lines indicate 200°F hot water (or 45°F chilled water) distribution piping (Secondary Loops). The square boxes represent transitions from 380°F transmission water through heat exchangers to 200°F water or to large absorptive air conditioning plants producing 45°F chilled water. Except for the base hospitals, a building may be supplied with TUS 200°F hot water <u>or</u> by TUS 45°F chilled water, but not with hot water and chilled water simultaneously. Thus a building with chilled water cooling in the summer must have year-round domestic hot water from electrically supplied heaters. For a TUS simulation on a given day, TDIST2 operates a given secondary loop in either the hot water or chilled water mode, but different secondary loops

may simultaneously be heating and cooling. This requirement may occur during winter-spring or spring-summer weather conditions. At these times, some secondary loops may be serving administration buildings requiring air conditioning while other secondary loops may be serving residential buildings requiring space heating. The simulation automatically operates secondary loops to supply the appropriate product.

The circles in Fig. 4.1 indicate load centers. For purposes of the simulation, several buildings in the vicinity of a load center are assumed to have their loads applied at the load center. The primary distribution system receives water at 380°F from the central power plant and distributes the water to the primary-to-secondary heat exchangers. Return water leaves these heat exchangers at 150°F and returns to the central station. The water temperature decreases along the length of the pipes as heat is lost to the ground. The TDIST2 simulation describes these energy losses.

The secondary systems produce water at 200°F in the primary-to-secondary heat exchangers, and distribute this water to the load centers (consumers). Water leaves the load centers at 80°F and returns to the primary-to-secondary heat exchangers. If cooling is required, the primary-to-secondary heat exchangers become absorptive air conditioning plants. The secondary loops then distribute

water at 45°F for chilled water air conditioning. Water leaves the load centers at 55°F and returns to the air conditioning plant. Once again, the TDIST2 simulation includes the effects of heat transfer from the ambient to the chilled water.

In the Fort Knox studies, computer simulations have been run for both design Options 1 and 2 for Thermal/Electrical load split values of 100%, 80%, 60%, 40%, 20% and 0%. These values span the complete range of thermal/electrical use for each TUS design option. Lifetime present-worth Total TES costs for each split value for each TUS option are also computed. Decreasing the thermal/electric split value from 100% to 80%, etc. is accomplished by shedding buildings from the 100% TUS until a total of 20% of the winter peak load has been transferred to electrical supply. Loops are shed selectively in decreasing order of a calculated energy density parameter. This parameter for a given loop is the peak winter load for the loop divided by the total length of piping required to supply the loop. Thus, the loops which are dropped first are those which supply the least power with the most piping. Using this criterion, the surviving heat exchanger and load center numbers for each split value are listed in Appendix C. Appendix C also lists the building distribution assignments; that is, the number of buildings of each type which are applied to the load centers in the TDIST2 simulation.

Pipe sizes for each TUS configuration are calculated by TDIST2 to give a maximum fluid velocity of 8 ft/sec at the design mass flow rate through a given pipe. This calculated pipe size is then compared to schedules of commercially available high pressure insulated pipe. The final pipe size selected is the smallest available pipe size which will give a fluid velocity of 8 ft/sec or less. Appendix C lists the pipes, lengths and pipe cross-sectional areas for each pipe for each TUS design configuration.

REFERENCES

1. Goldman, S.B., Best, F.R., Golay, M.W., "TDIST2, A Computer Program for Community Energy Consumption Analysis and Total Energy System Design," Project Report, Contract No. DAAK02-74-C-0308, Department of Nuclear Engineering, MIT, June 1977.

2. Stetkar, J.S., Golay, M.W., "TDIST, A Program for Community Energy Demand Analysis and Total Energy System Response Simulation," Project Report, Contract No. DAAK02-74-C-0308, Department of Nuclear Engineering, MIT, August 1976.

3. Stetkar, J.S., Golay, M.W., Best, F.R., "Design of a Nuclear Powered Total Energy System for Ft. Bragg, North Carolina," Project Report, Contract No. DAAK02-74-C-0308, Department of Nuclear Engineering, MIT, May 1976.

CHAPTER 5

FORT KNOX UTILITY SYSTEM SIMULATION RESULTS

In the preceding two chapters, the fifteen Fort Knox energy consumer categories and the layout of the thermal utility system options are presented. The basic data required to determine the optimal TES design — affording the minimum total cost over its lifetime — are the installed power plant thermal and electrical power generation capacities, the thermal energy storage reservoir capacity, the energy loss rates and size of the Thermal Utility System (TUS), and the total energy required annually by the power plant. In order to determine these economic study input data and to investigate the behavior of the TUS over a range of seasonally-varying thermal and electrical energy demand schedules, many computer simulations, each covering a 24-hour period, have been performed. These calculations involve varying the energy supply system options for the base during six different weather days throughout the year. It has been assumed that the seasoned weather conditions at Fort Knox are approximately symmetrical between the spring and autumn. The days chosen for study have been designated as the peak winter heating demand day, an average winter day, an early spring day (identical to a late fall day), a late spring (early fall) day, an average summer day, and the peak summer cooling demand day. The sequence of simula-

tion runs is as follows:

1. Run case of peak winter design hour in order to size pipes and heat exchangers for winter loads.
2. Run case of peak summer design hour in order to size pipes for summer loads.
3. Compare winter peak and summer peak pipe sizes. If absorptive air conditioning is specified, secondary loop pipe sizes will usually be larger in summer than in winter (due to lower available ΔT, 10°F in summer but 120°F in winter). Select the largest pipe sizes from both simulations to form the "design" pipe configuration.
4. Using the "design" piping, run the cases of the six seasonal days in order to determine component sizing and the annual energy consumption rate.

5.1 Daily Energy Demand Schedules

Figures 5.1-5.34 display the results of the winter peak and summer peak design simulations for the complete range of thermal/electrical split values for both the absorptive and compressive air conditioning options. Also shown are the seasonal simulation runs for the 100%, 60% and 0% split values for the absorptive air conditioning option. The figures show the diurnal variation of electrical demand, TUS thermal power demand, the hot mass stored in the reservoir and the instantaneous power being delivered to

the reservoir. The simulations are based on nuclear power station electrical generation efficiency of 38%.

Generally, in the plots, the peak heating power demand occurs in the interval 1:00 to 3:00 AM at the coldest time of the night and before the sunrise. The peak cooling demand usually occurs in the early afternoon. Note that the hot mass in the reservoir decreases when the thermal demand exceeds the power plant thermal output to the reservoir, and that the hot mass in the reservoir increases when thermal demand is less than power plant thermal output to the reservoir. The computer model of plant operation is based on an assumption of constant thermal power operation, typically required for nuclear power plants. That is, the nuclear reactor core thermal power remains constant over the course of a day, but is allowed to vary seasonally. Thus core power output on the peak winter design day might be a constant 150 MW(t), and on the typical spring-summer day it would be a constant 70 MW(t) (although the core would be capable of producing 150 MW(t)). Under these operating conditions the waste heat available as a result of required electrical energy generation is stored in the thermal reservoir or is dissipated in the plant's cooling tower. If more thermal energy is required over the course of the day than the net energy available from electrical waste heat, the simulation automatically increases reactor power to supply the total thermal energy demand. Typically, during

the winter at high thermal/electrical split values, core power will be increased over that required solely for electrical production in order to meet the thermal demand. However, in the summer or at low thermal/electrical split values, the waste heat available from electrical production is larger than the thermal demand and so excess energy is dissipated in the cooling tower. The computer simulation automatically calculates the minimum required reactor size and reservoir volume necessary to meet the day's thermal and electrical demand schedule. Note that the required reservoir volume depends not only on the magnitude of the thermal demand, but also on the difference between the thermal demand and plant thermal output. Therefore, as the figures show, the required reservoir volume for the absorptive air conditioning option is determined by the summertime mismatch of thermal and electrical power schedules, while for the compressive air conditioning option the winter thermal-electrical power schedule mismatch determines the required reservoir volume.

The reservoir mass and thermal energy flows are not shown for the 0% thermal/electrical split case since no TUS is considered to exist in this all-electric case. For some split values in the winter-spring or spring-summer cases, the thermal demand is seen to go to zero. This indicates that due to mild weather conditions the space conditioning demand is negligibly small and the TUS has therefore been turned off.

AD-A043 701 MASSACHUSETTS INST OF TECH CAMBRIDGE DEPT OF NUCLEAR--ETC F/G 10/2
ANALYSIS OF NUCLEAR AND COAL FUELED TOTAL ENERGY SYSTEM OPTIONS--ETC(U)
JUN 77 F R BEST, S B GOLDMAN, M W GOLAY DAAK02-74-C-0308

UNCLASSIFIED USAFESA-RT-2039 NL

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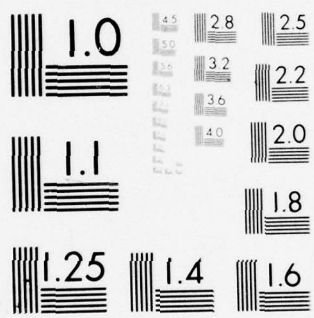


MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS-1963-A

Figure 5.1

TES PARAMETERS - FORT KNOX

100% TUS ABS A/C PEAK WINTER

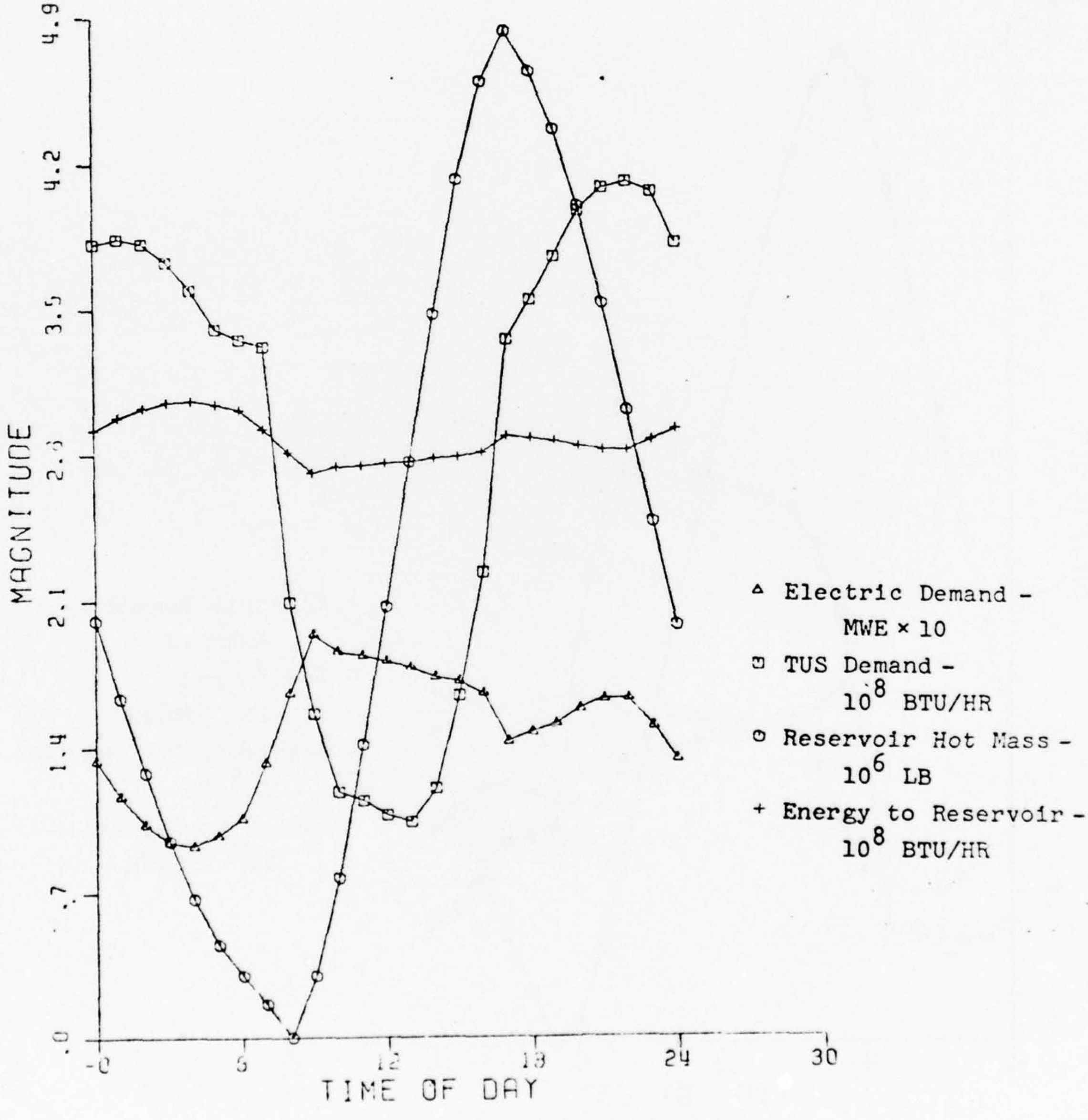


Figure 5.2

TES PARAMETERS - FORT KNOX

100% TUS ABS A/C PEAK SUMMER

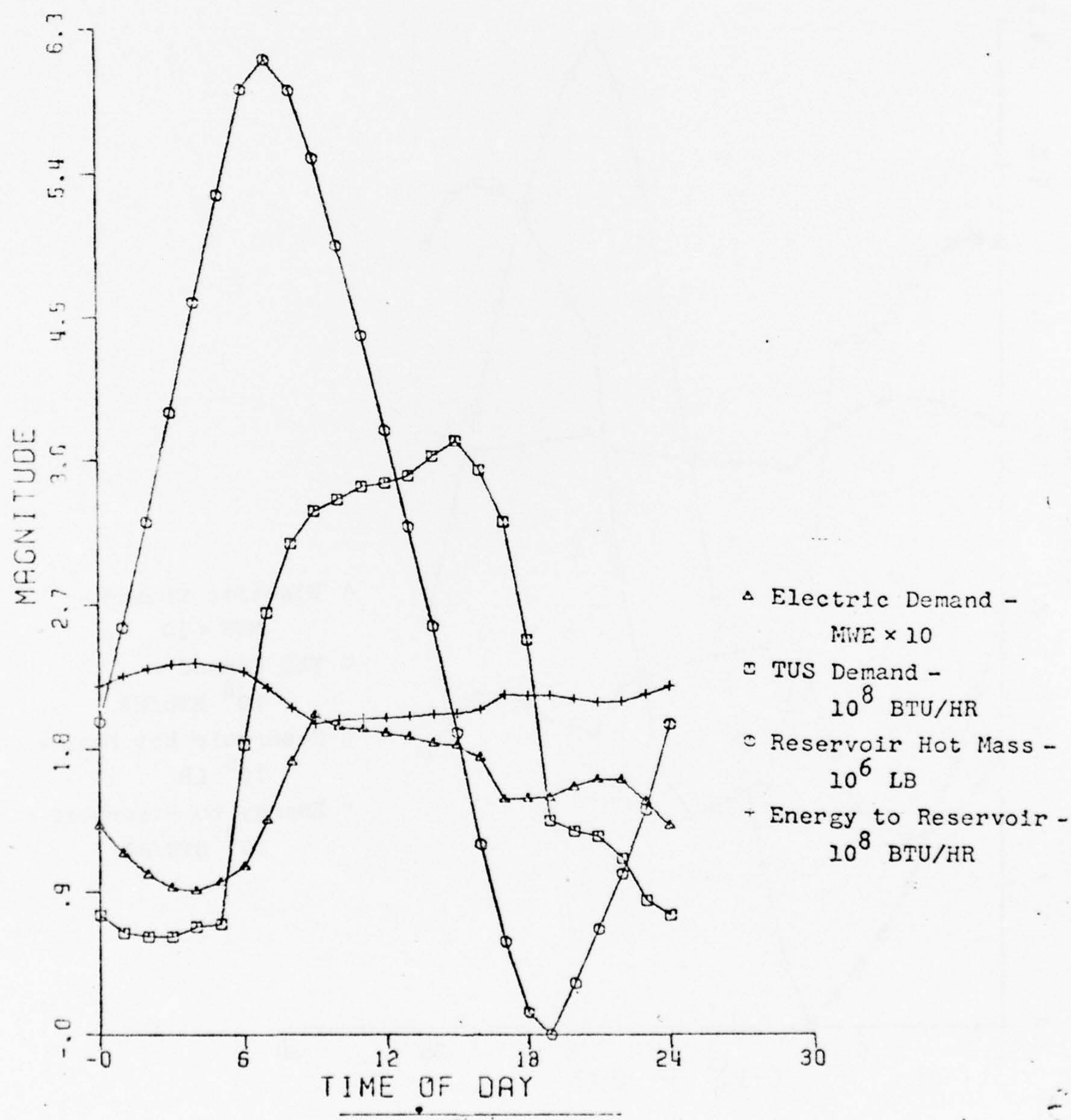
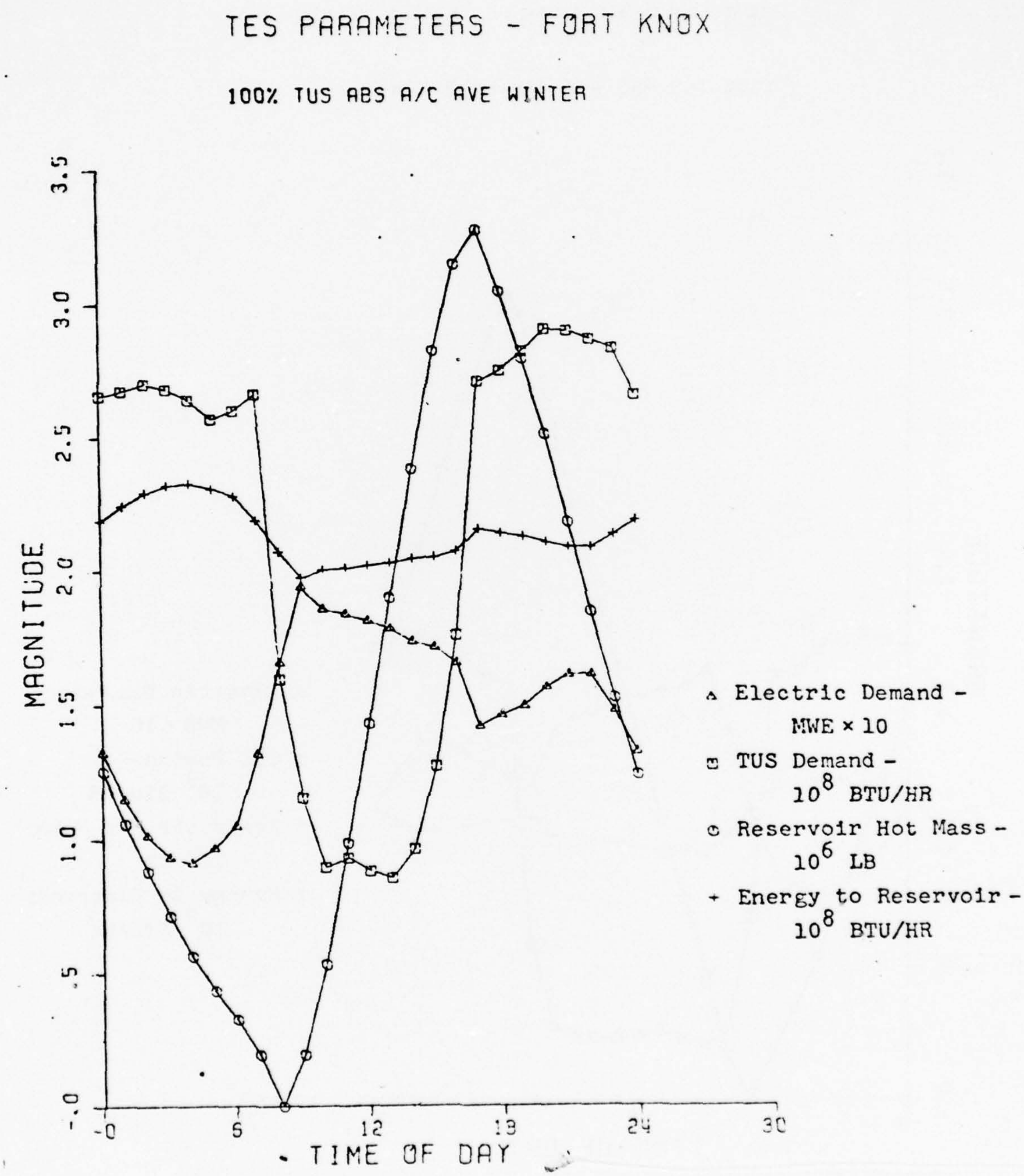


Figure 5.3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100% TUS ABS A/C AVE WINTER
MAGNITUDE
3.5
3.0
2.5
2.0
1.5
1.0
.5
-.0
-0
6
12
18
24
30
TIME OF DAY
Electric Demand - MWE × 10
TUS Demand - 10^8 BTU/HR
Reservoir Hot Mass - 10^6 LB
Energy to Reservoir - 10^8 BTU/HR

Figure 5.4

TES PARAMETERS - FORT KNOX

100% TUS ABS A/C WINTER-SPRING

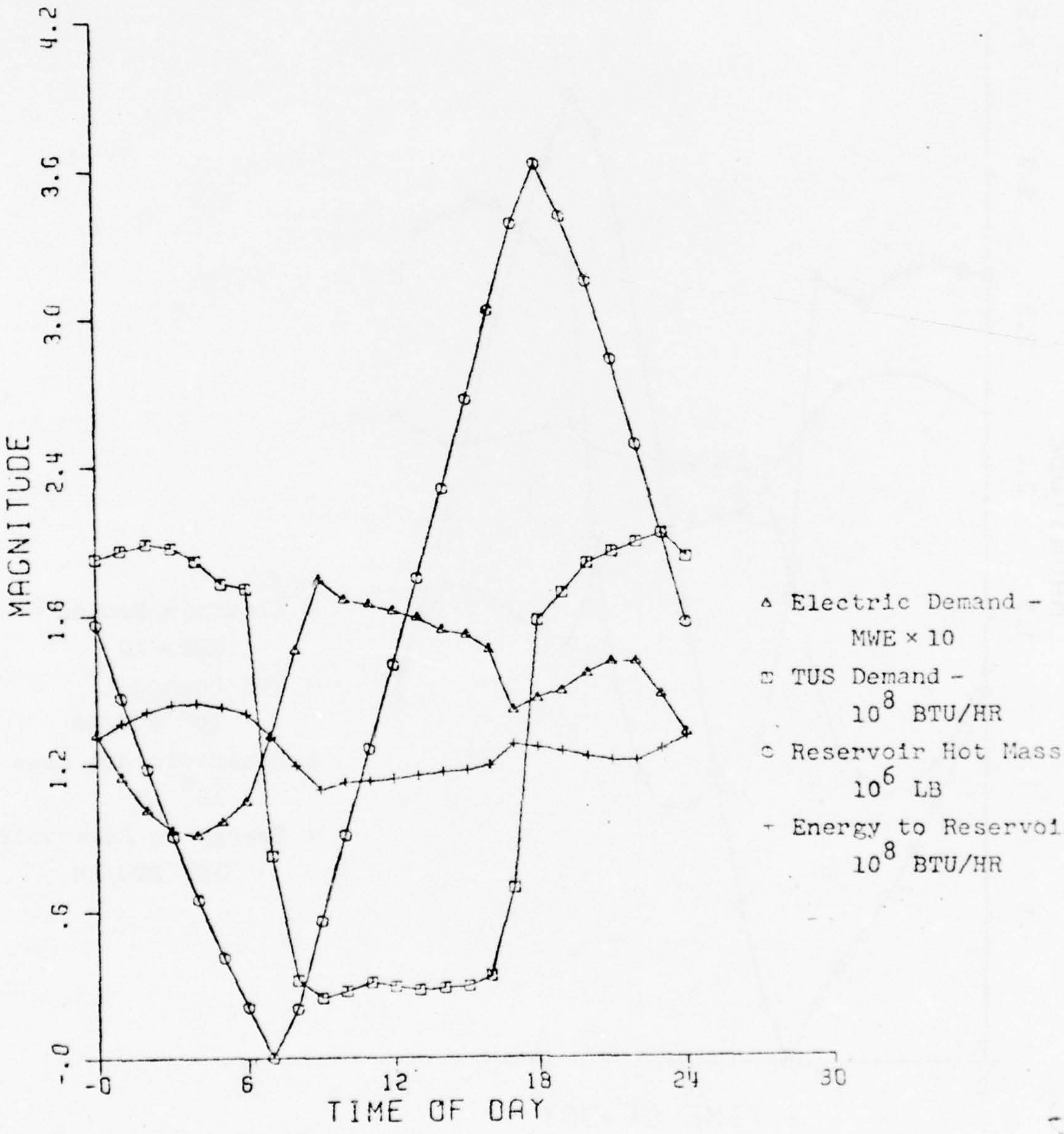


Figure 5.5

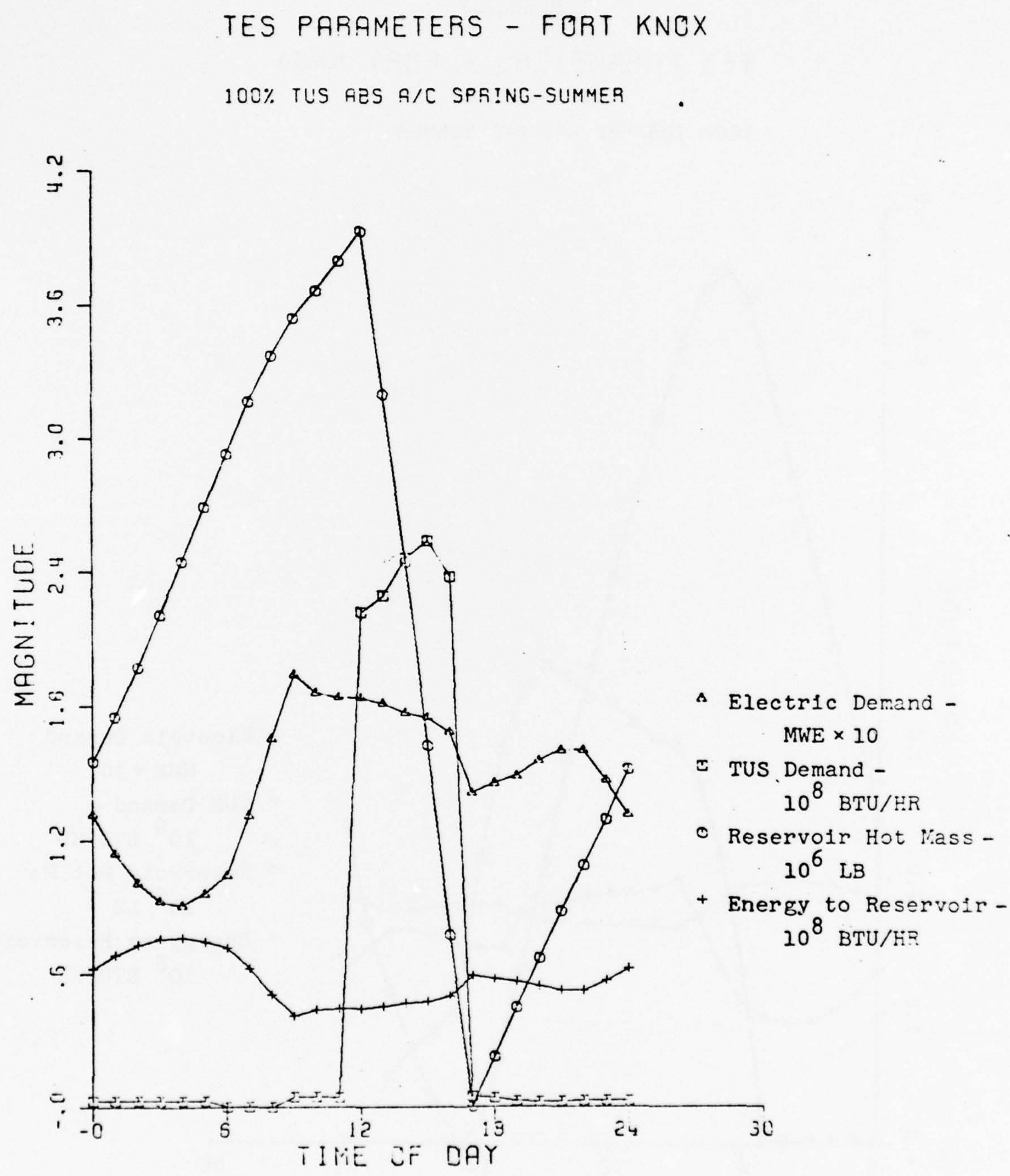

TES PARAMETERS - FØRT KNOX
100% TUS ABS A/C SPRING-SUMMER
MAGNITUDE
-.0
.6
1.2
1.6
2.4
3.0
3.6
4.2
TIME OF DAY
-0
6
12
18
24
30
Electric Demand - MWE × 10
TUS Demand - 10^8 BTU/HR
Reservoir Hot Mass - 10^6 LB
Energy to Reservoir - 10^8 BTU/HR

Figure 5.6

TES PARAMETERS - FORT KNOX

100% TUS ABS A/C AVE SUMMER

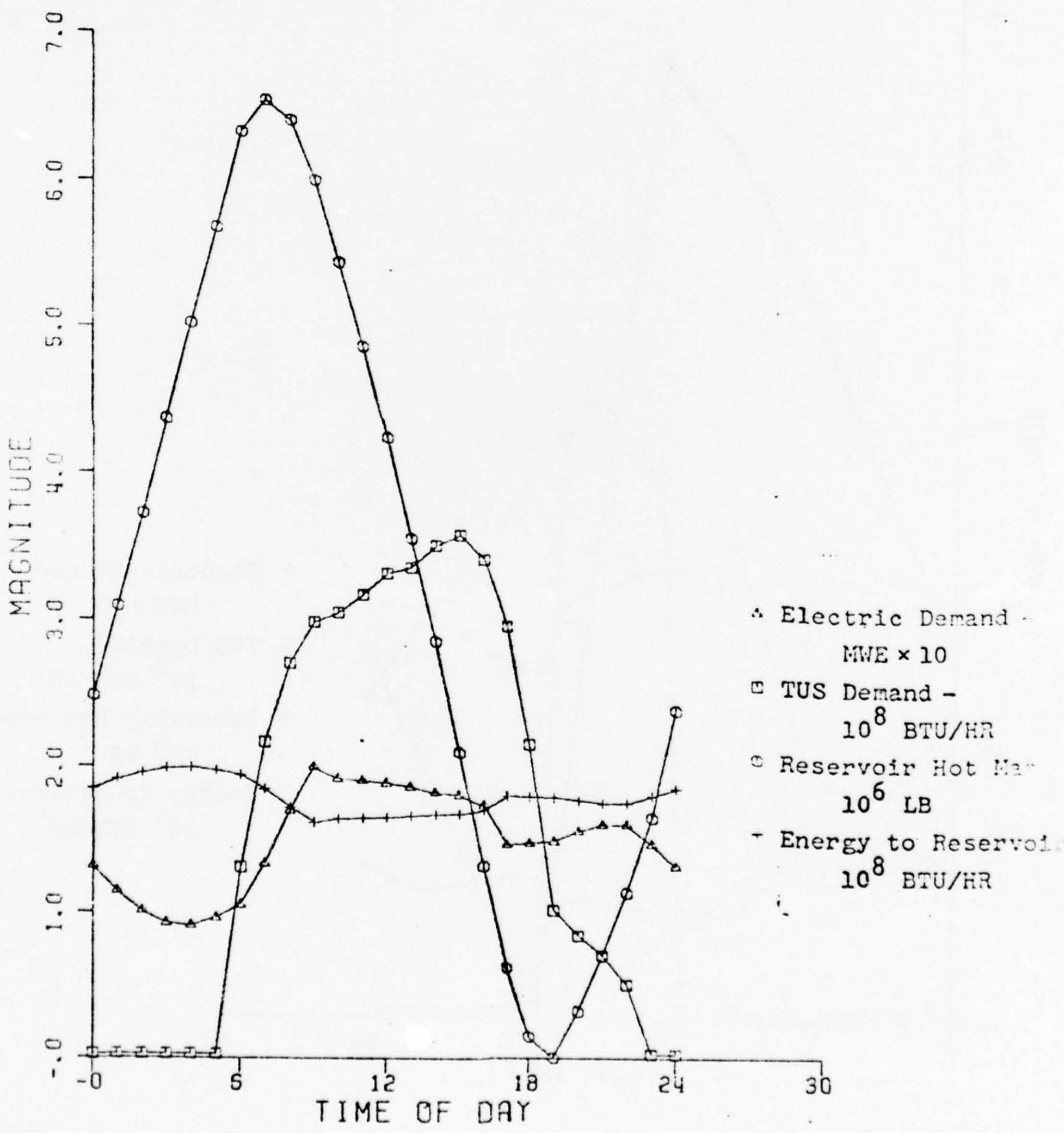
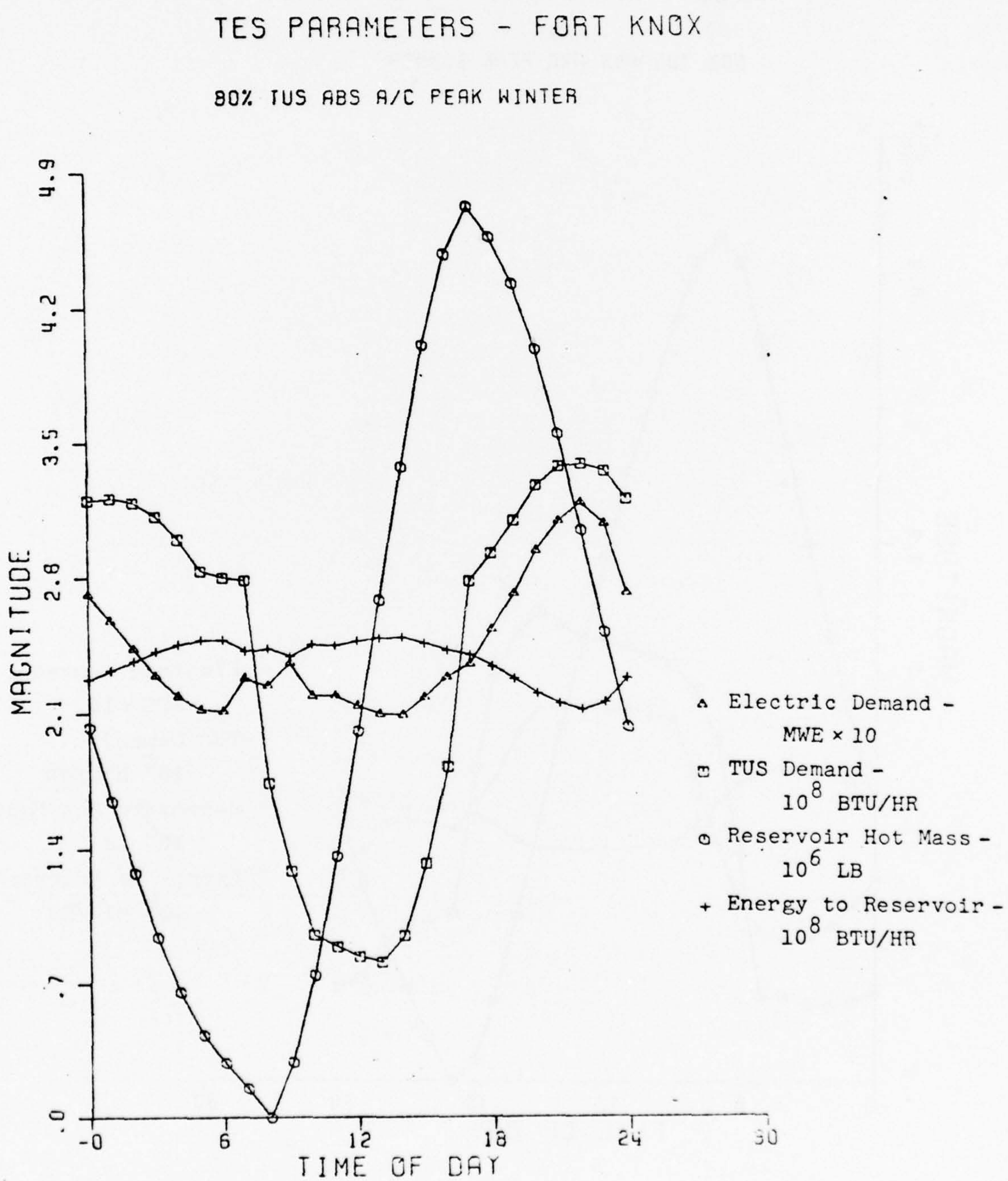


Figure 5.7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80% TUS ABS A/C PEAK WINTER
MAGNITUDE
4.9
4.2
3.5
2.8
2.1
1.4
.7
.0
-0
6
12
18
24
30
TIME OF DAY
Electric Demand - MWE × 10
TUS Demand - 10^8 BTU/HR
Reservoir Hot Mass - 10^6 LB
Energy to Reservoir - 10^8 BTU/HR

Figure 5.8

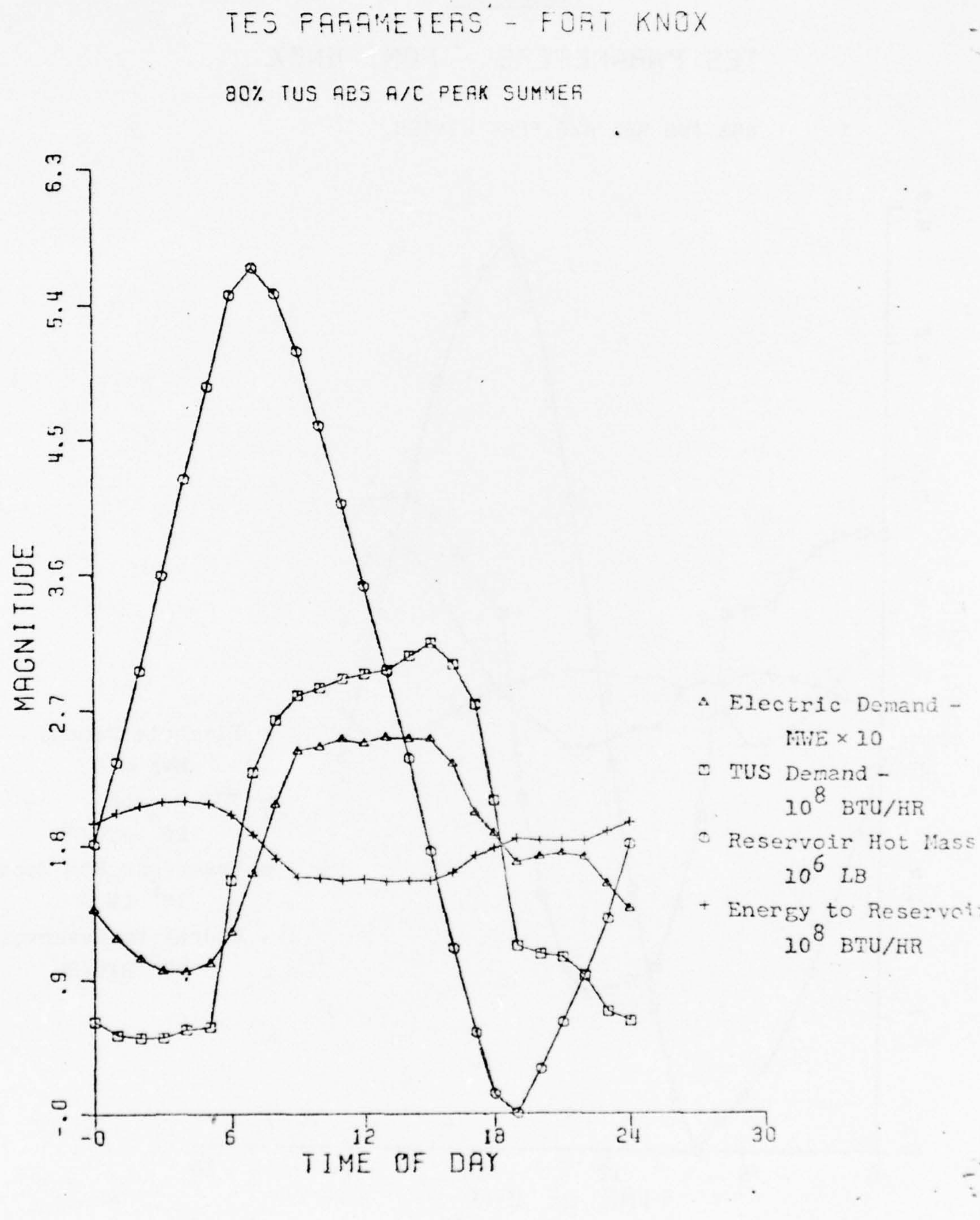
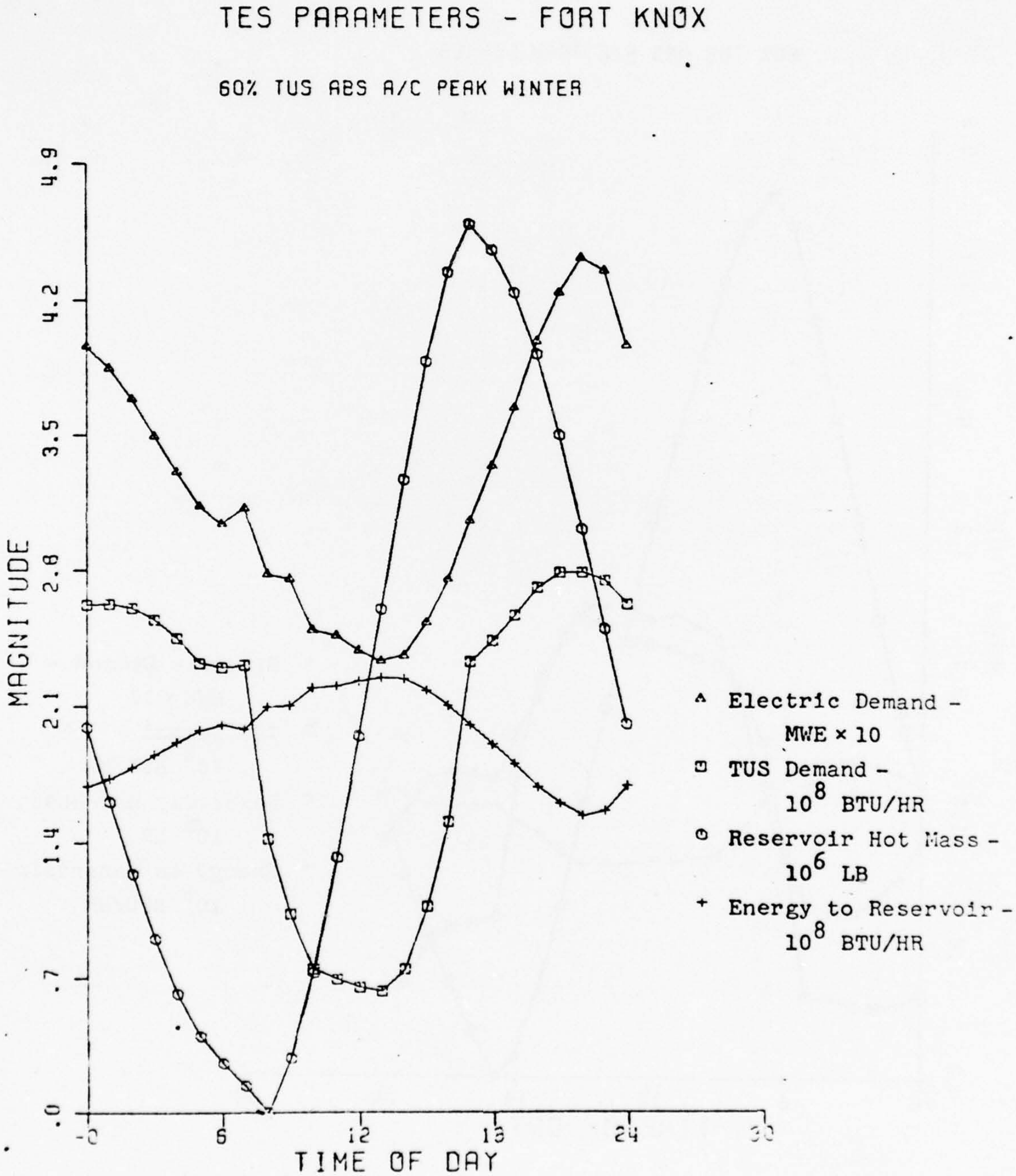

TES PARAMETERS - FORT KNOX
80% TUS ABS A/C PEAK SUMMER
MAGNITUDE
6.3
5.4
4.5
3.6
2.7
1.8
.9
-.0
-0
6
12
18
24
30
TIME OF DAY
Electric Demand - MWE × 10
TUS Demand - 10^8 BTU/HR
Reservoir Hot Mass 10^6 LB
Energy to Reservoi: 10^8 BTU/HR

Figure 5.9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60% TUS ABS A/C PEAK WINTER
MAGNITUDE
4.9
4.2
3.5
2.8
2.1
1.4
.7
.0
-0
6
12
18
24
30
TIME OF DAY
Electric Demand - MWE × 10
TUS Demand - 10^8 BTU/HR
Reservoir Hot Mass - 10^6 LB
Energy to Reservoir - 10^8 BTU/HR

Figure 5.10

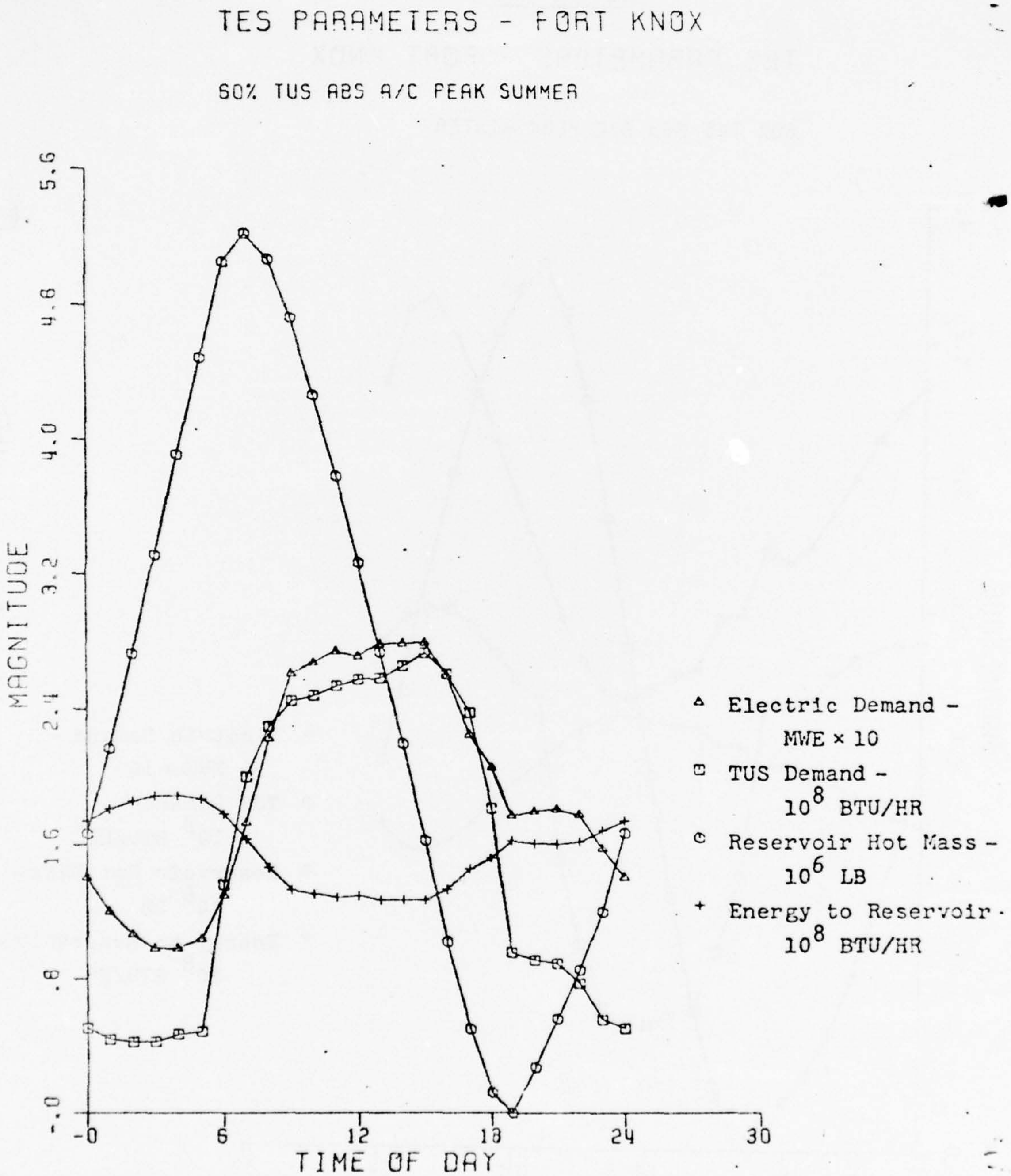

TES PARAMETERS - FORT KNOX
60% TUS ABS A/C PEAK SUMMER
MAGNITUDE
5.6
4.8
4.0
3.2
2.4
1.6
.8
-.0
-0
6
12
18
24
30
TIME OF DAY
Electric Demand - MWE × 10
TUS Demand - 10^8 BTU/HR
Reservoir Hot Mass - 10^6 LB
Energy to Reservoir - 10^8 BTU/HR

Figure 5.11

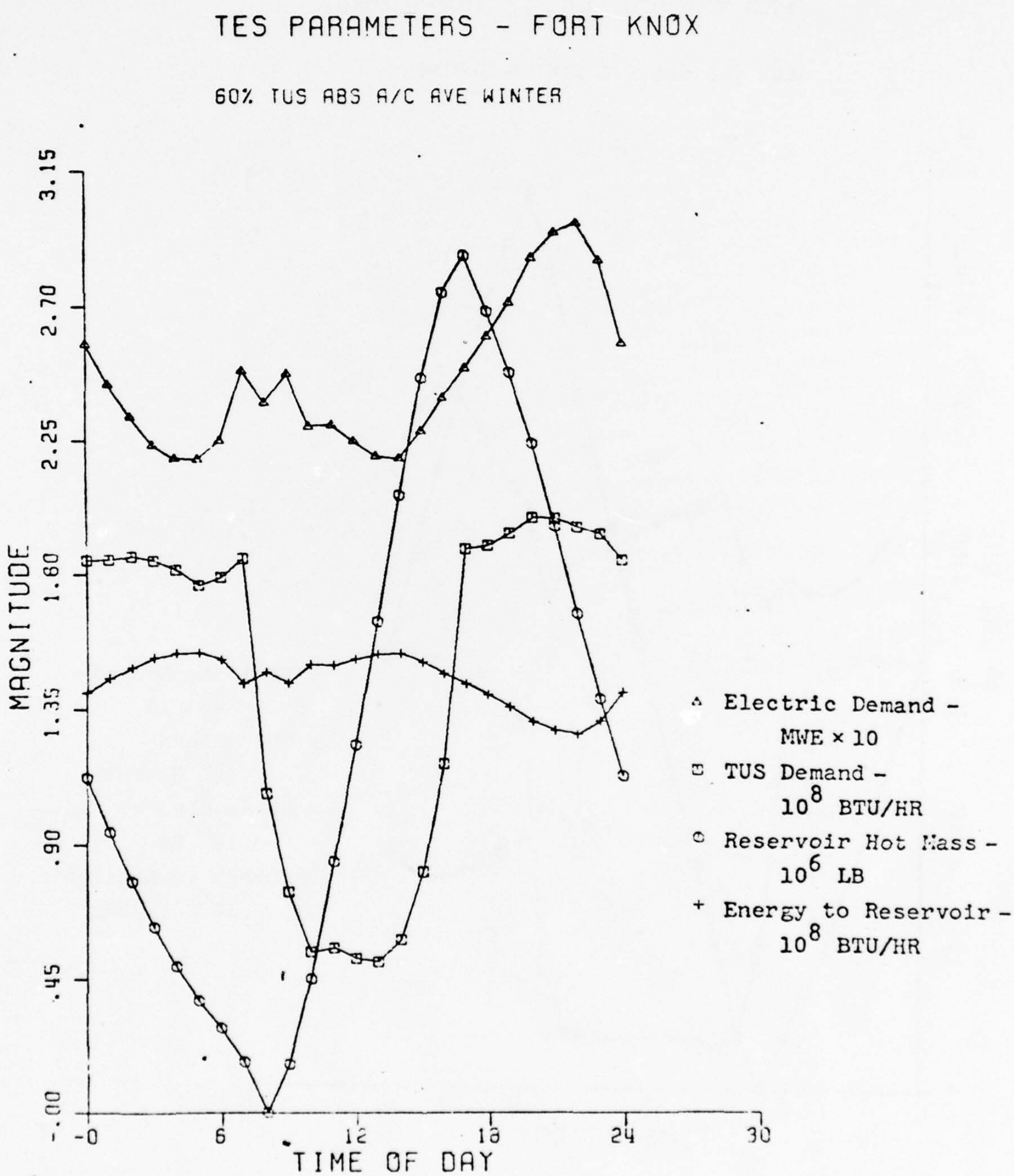

TES PARAMETERS - FORT KNOX
60% TUS ABS A/C AVE WINTER
MAGNITUDE
3.15
2.70
2.25
1.80
1.35
.90
.45
-.00
-0
6
12
18
24
30
TIME OF DAY
Electric Demand - MWE × 10
TUS Demand - 10^8 BTU/HR
Reservoir Hot Mass - 10^6 LB
Energy to Reservoir - 10^8 BTU/HR

Figure 5.12

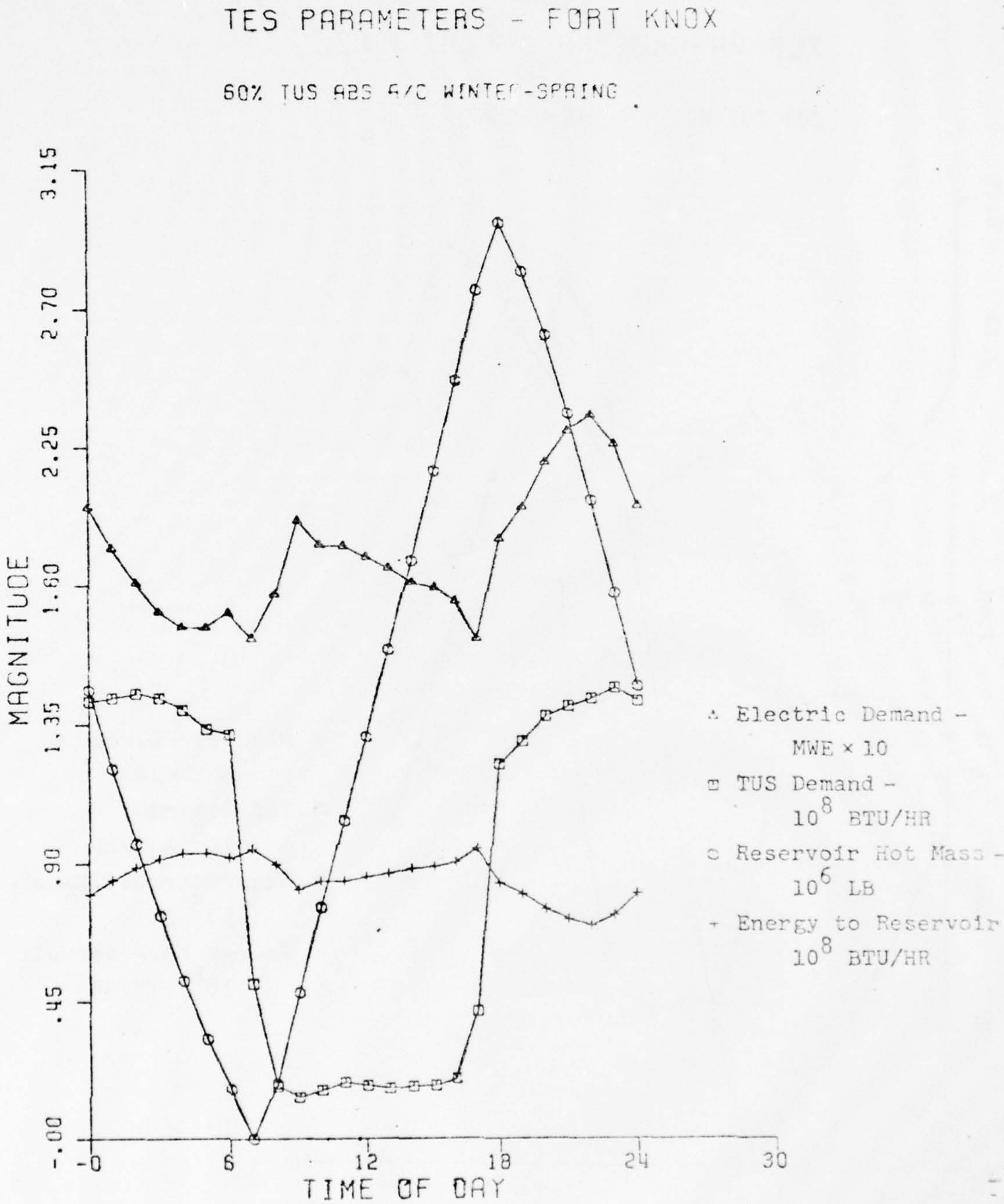
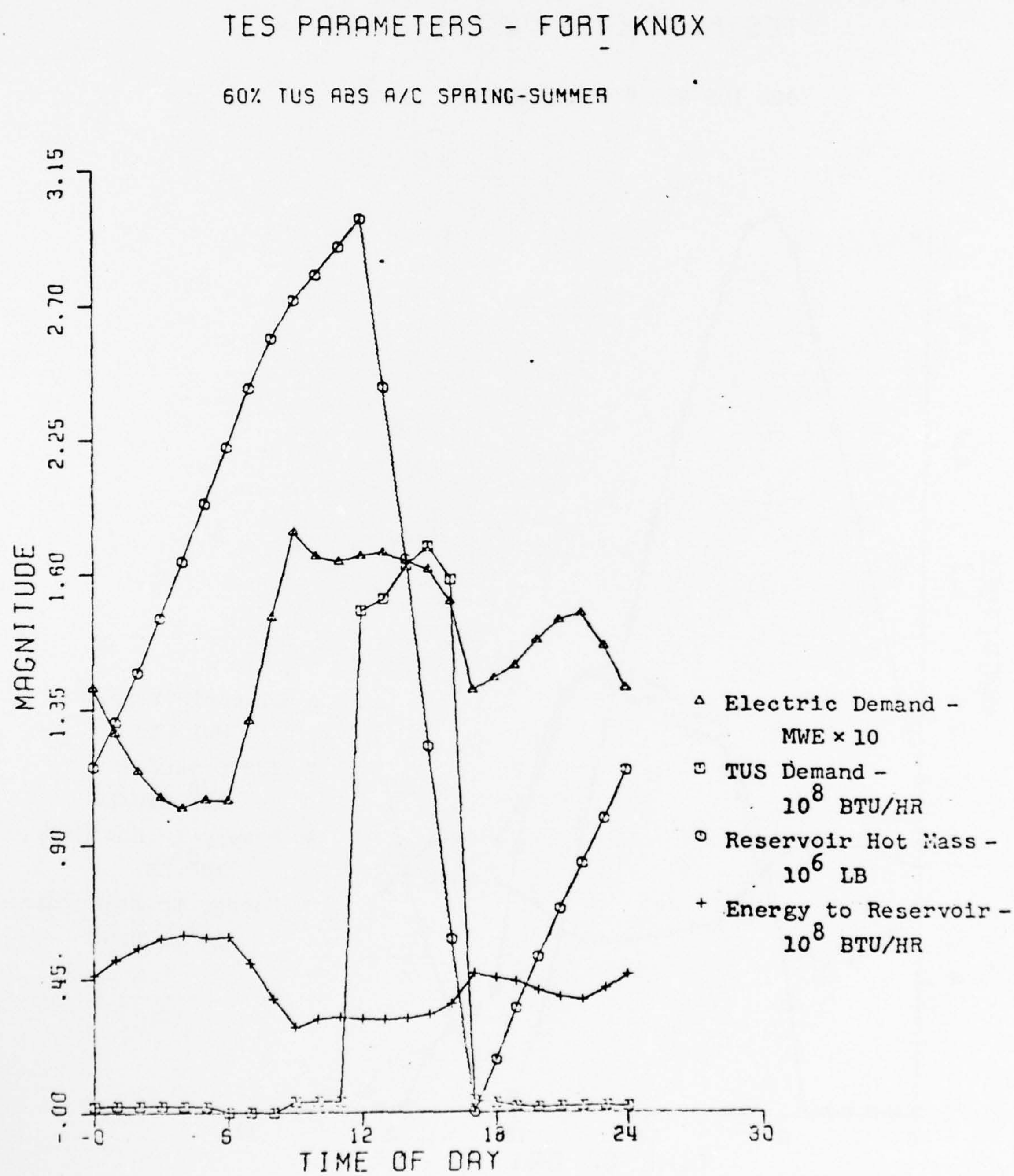

TES PARAMETERS - FORT KNOX
60% TUS ABS A/C WINTER-SPRING
MAGNITUDE
3.15
2.70
2.25
1.80
1.35
.90
.45
-.00
-0
6
12
18
24
30
TIME OF DAY
Electric Demand - MWE × 10
TUS Demand - 10^8 BTU/HR
Reservoir Hot Mass - 10^6 LB
Energy to Reservoir 10^8 BTU/HR

Figure 5.13



TES PARAMETERS - FORT KNOX
60% TUS ABS A/C SPRING-SUMMER
3.15
2.70
2.25
1.80
1.35
.90
.45
-.00
MAGNITUDE
-0
6
12
18
24
30
TIME OF DAY
Electric Demand - MWE × 10
TUS Demand - 10^8 BTU/HR
Reservoir Hot Mass - 10^6 LB
Energy to Reservoir - 10^8 BTU/HR

Figure 5.14

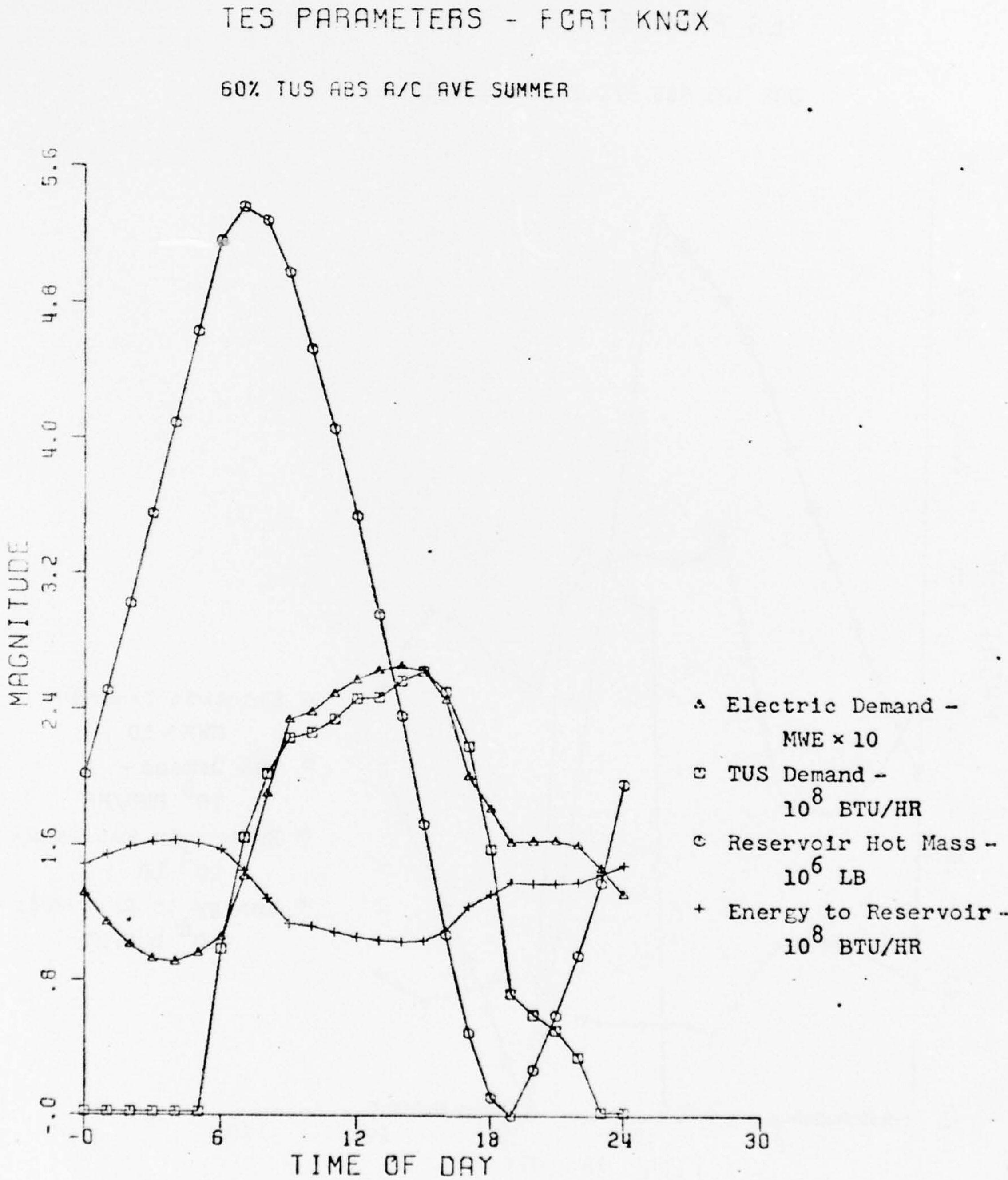

TES PARAMETERS - FORT KNOX
60% TUS ABS A/C AVE SUMMER
MAGNITUDE
-.0
.8
1.6
2.4
3.2
4.0
4.8
5.6
-0
6
12
18
24
30
TIME OF DAY
Electric Demand - MWE × 10
TUS Demand - 10^8 BTU/HR
Reservoir Hot Mass - 10^6 LB
Energy to Reservoir - 10^8 BTU/HR

Figure 5.15

TES PARAMETERS - FORT KNOX

40% TUS ABS A/C PEAK WINTER

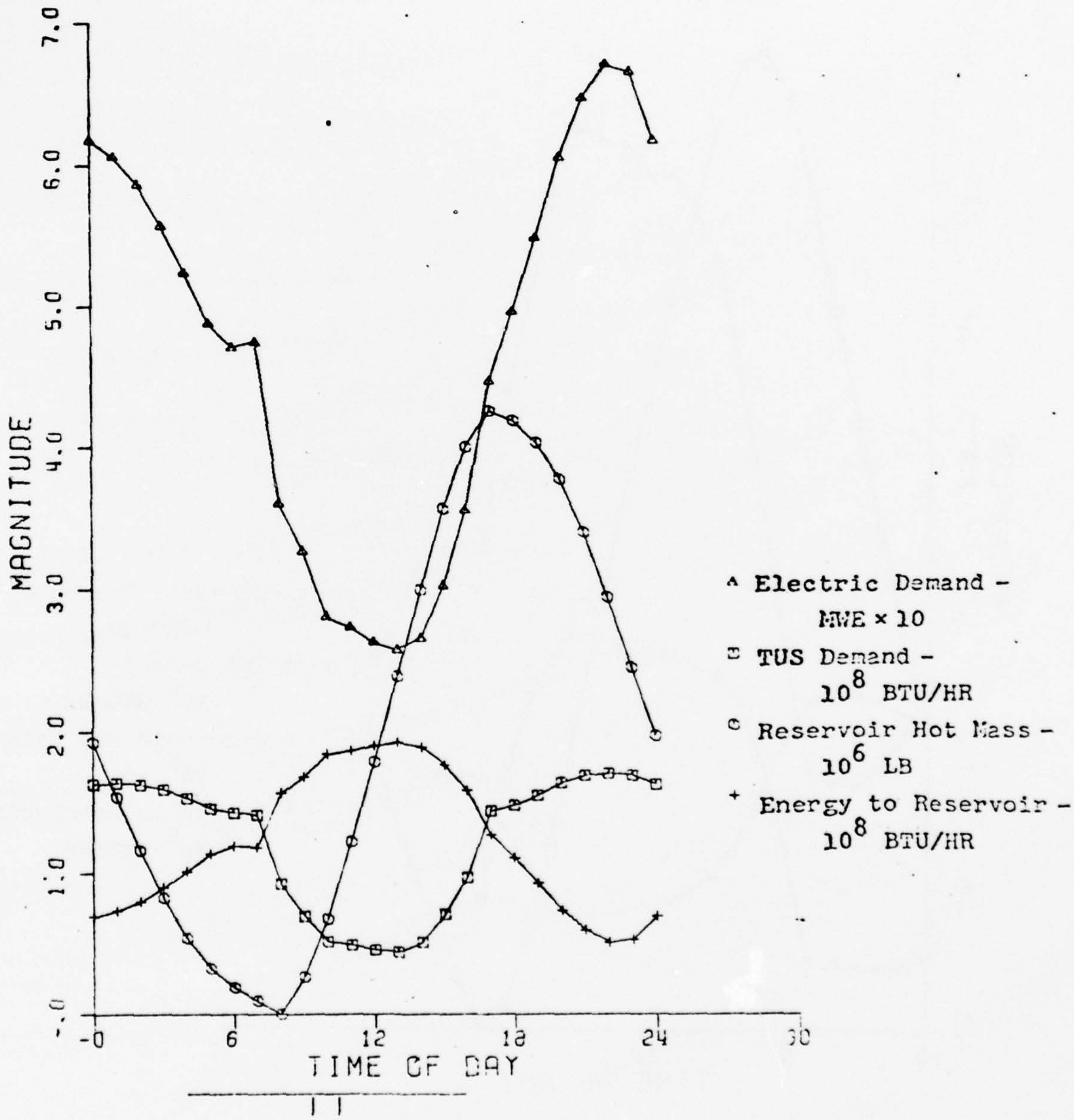


Figure 5.16

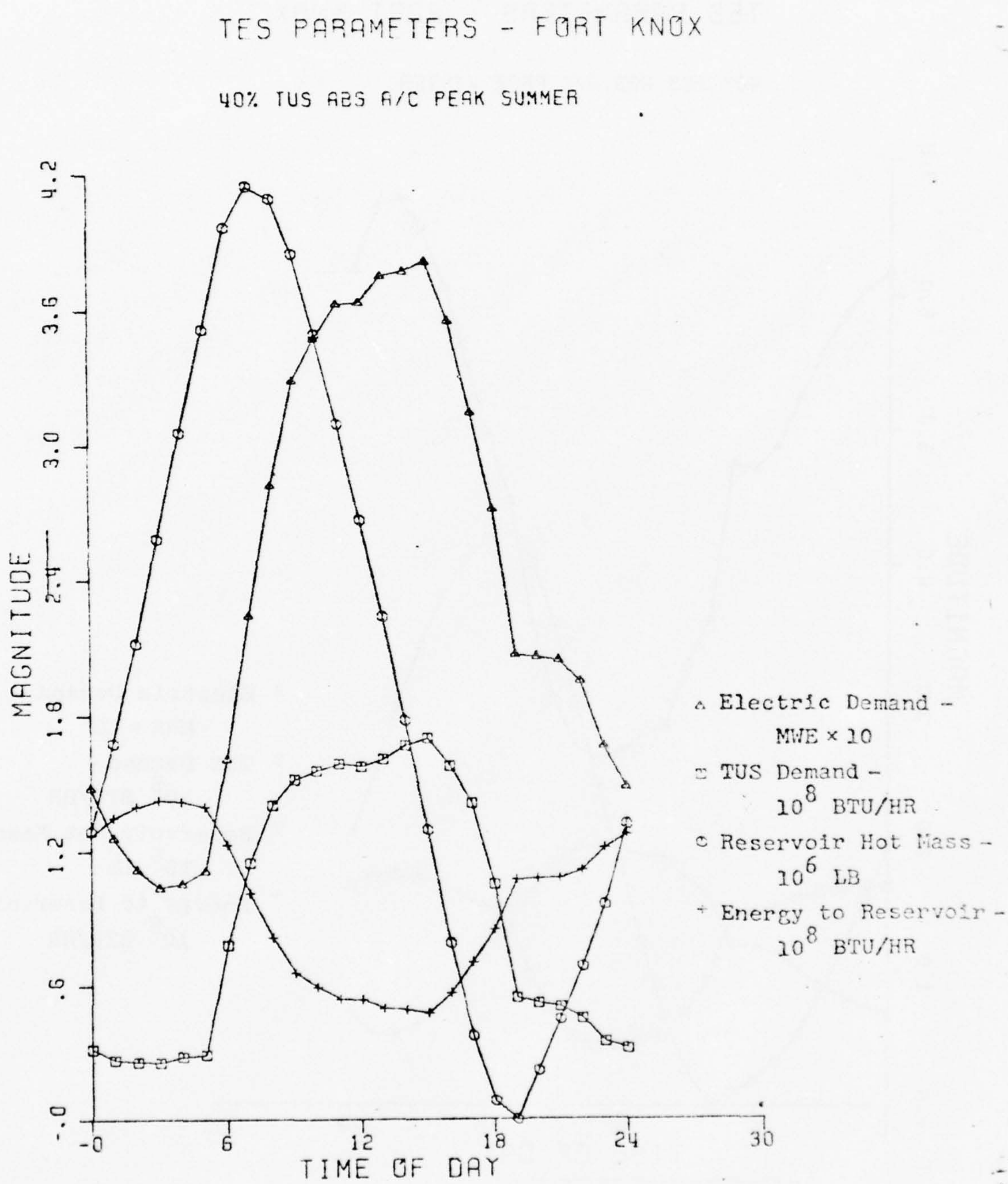

TES PARAMETERS - FORT KNOX
40% TUS ABS A/C PEAK SUMMER
MAGNITUDE
4.2
3.6
3.0
2.4
1.8
1.2
.6
-.0
-0
6
12
18
24
30
TIME OF DAY
△ Electric Demand - MWE × 10
□ TUS Demand - 10^8 BTU/HR
○ Reservoir Hot Mass - 10^6 LB
+ Energy to Reservoir - 10^8 BTU/HR

Figure 5.17

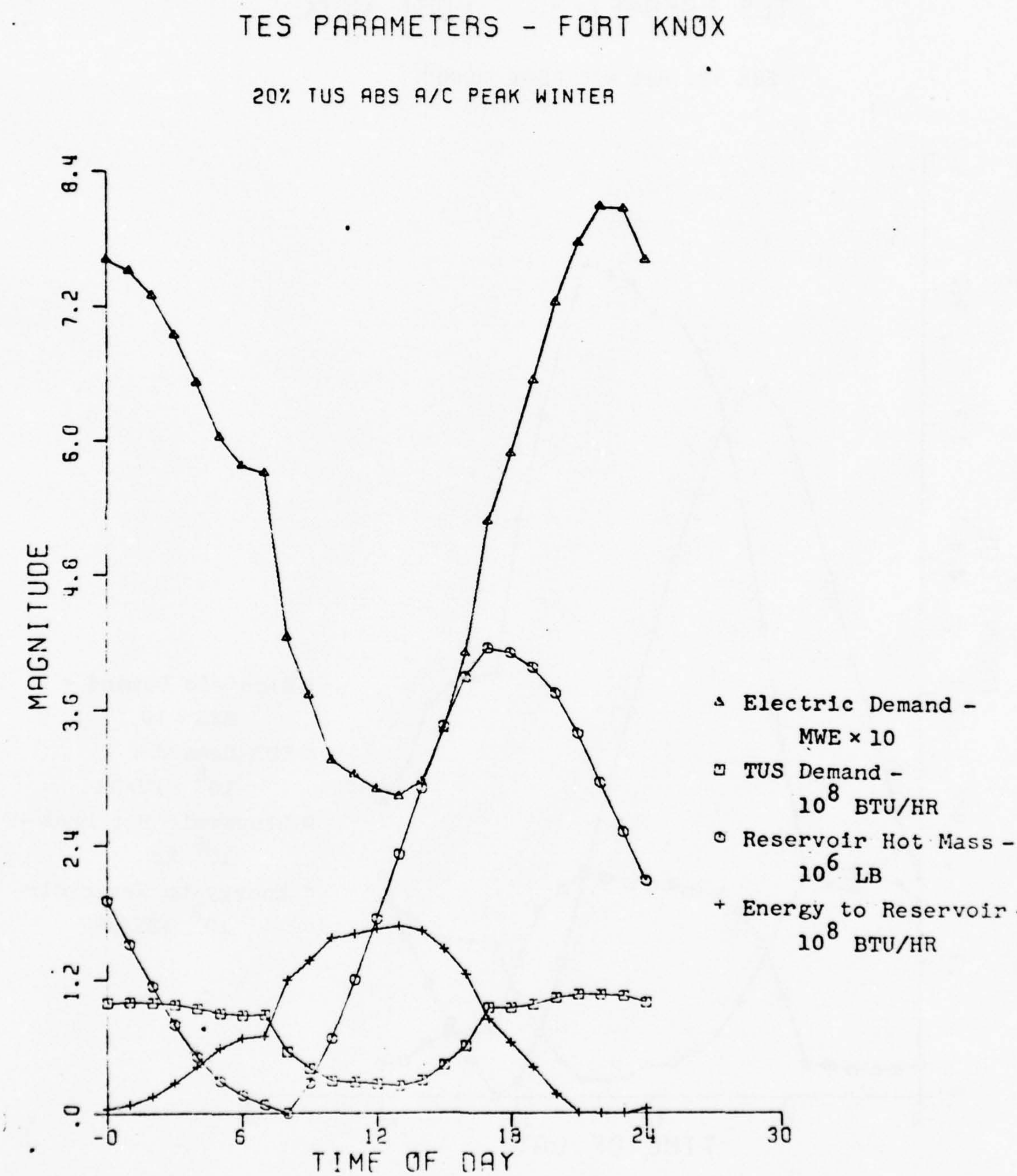

TES PARAMETERS - FORT KNOX
20% TUS ABS A/C PEAK WINTER
MAGNITUDE
.0
1.2
2.4
3.6
4.8
6.0
7.2
8.4
-0
6
12
18
24
30
TIME OF DAY
Electric Demand - MWE × 10
TUS Demand - 10^8 BTU/HR
Reservoir Hot Mass - 10^6 LB
Energy to Reservoir - 10^8 BTU/HR

Figure 5.18

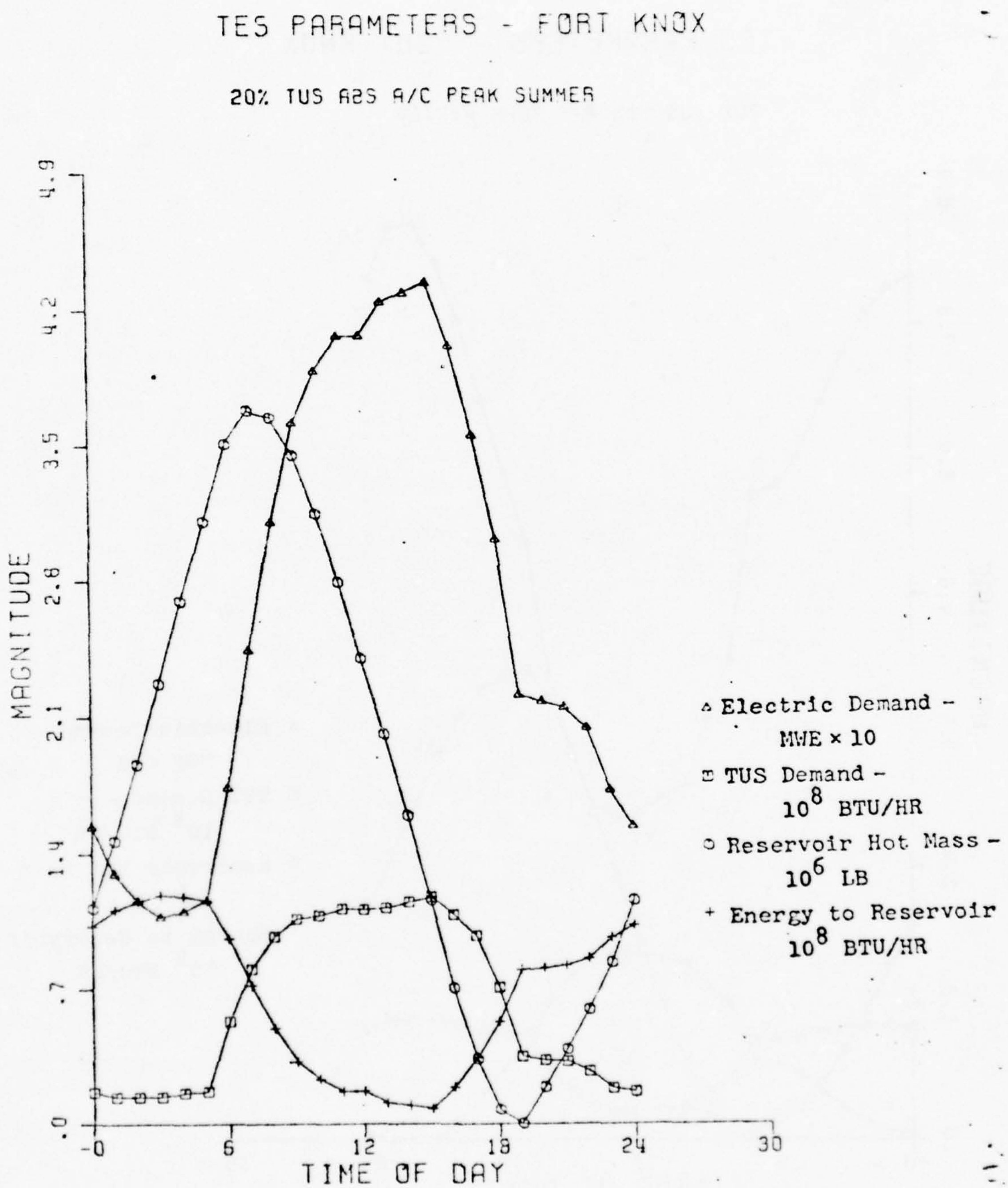

TES PARAMETERS - FORT KNOX
20% TUS A2S A/C PEAK SUMMER
MAGNITUDE
.0
.7
1.4
2.1
2.8
3.5
4.2
4.9
-0
6
12
18
24
30
TIME OF DAY
△ Electric Demand - MWE × 10
⊡ TUS Demand - 10^8 BTU/HR
○ Reservoir Hot Mass - 10^6 LB
+ Energy to Reservoir 10^8 BTU/HR

Figure 5.19

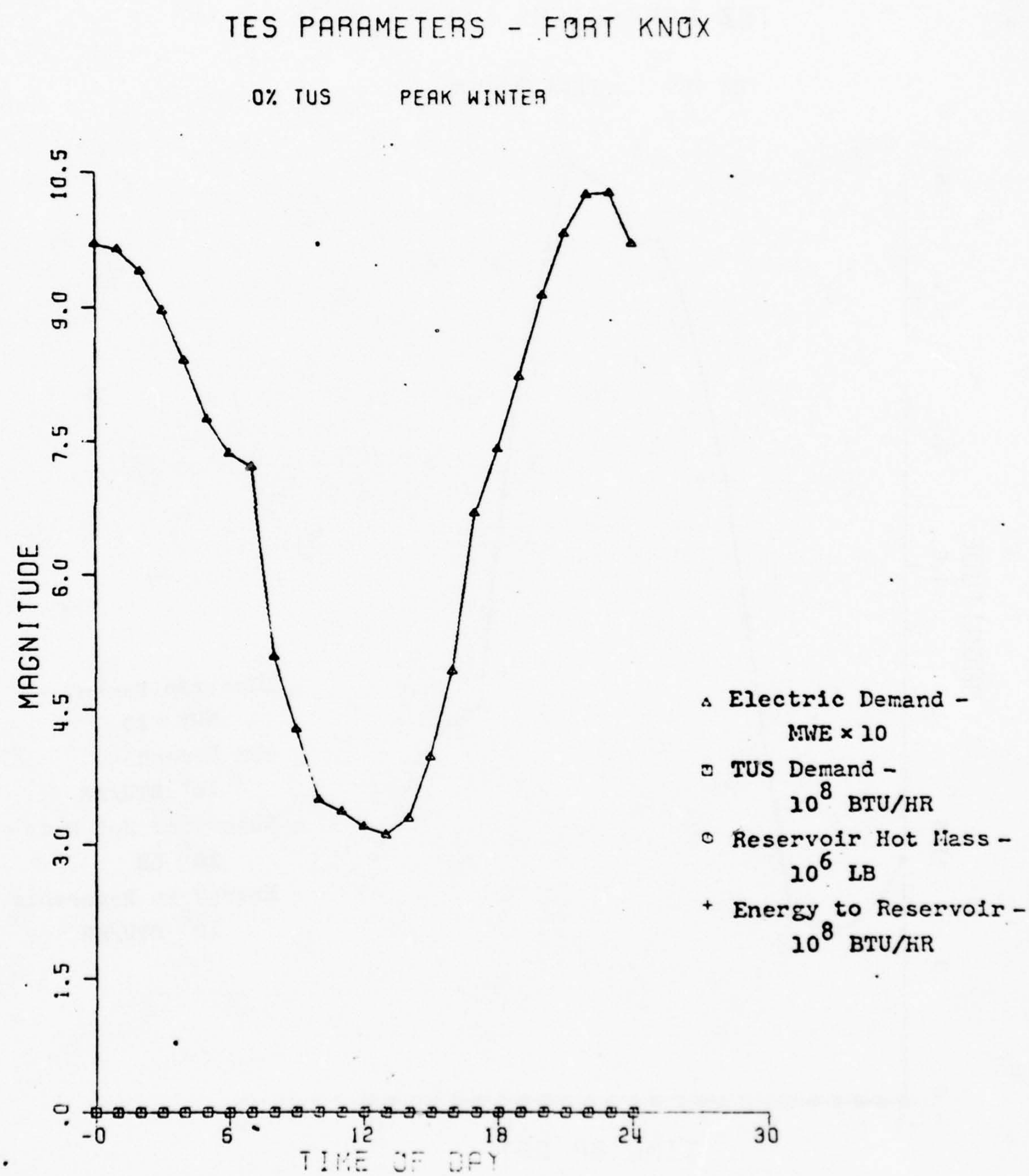

TES PARAMETERS - FORT KNOX
0% TUS PEAK WINTER
MAGNITUDE
10.5
9.0
7.5
6.0
4.5
3.0
1.5
.0
-0
6
12
18
24
30
TIME OF DAY
Electric Demand - MWE × 10
TUS Demand - 10^8 BTU/HR
Reservoir Hot Mass - 10^6 LB
Energy to Reservoir - 10^8 BTU/HR

Figure 5.20

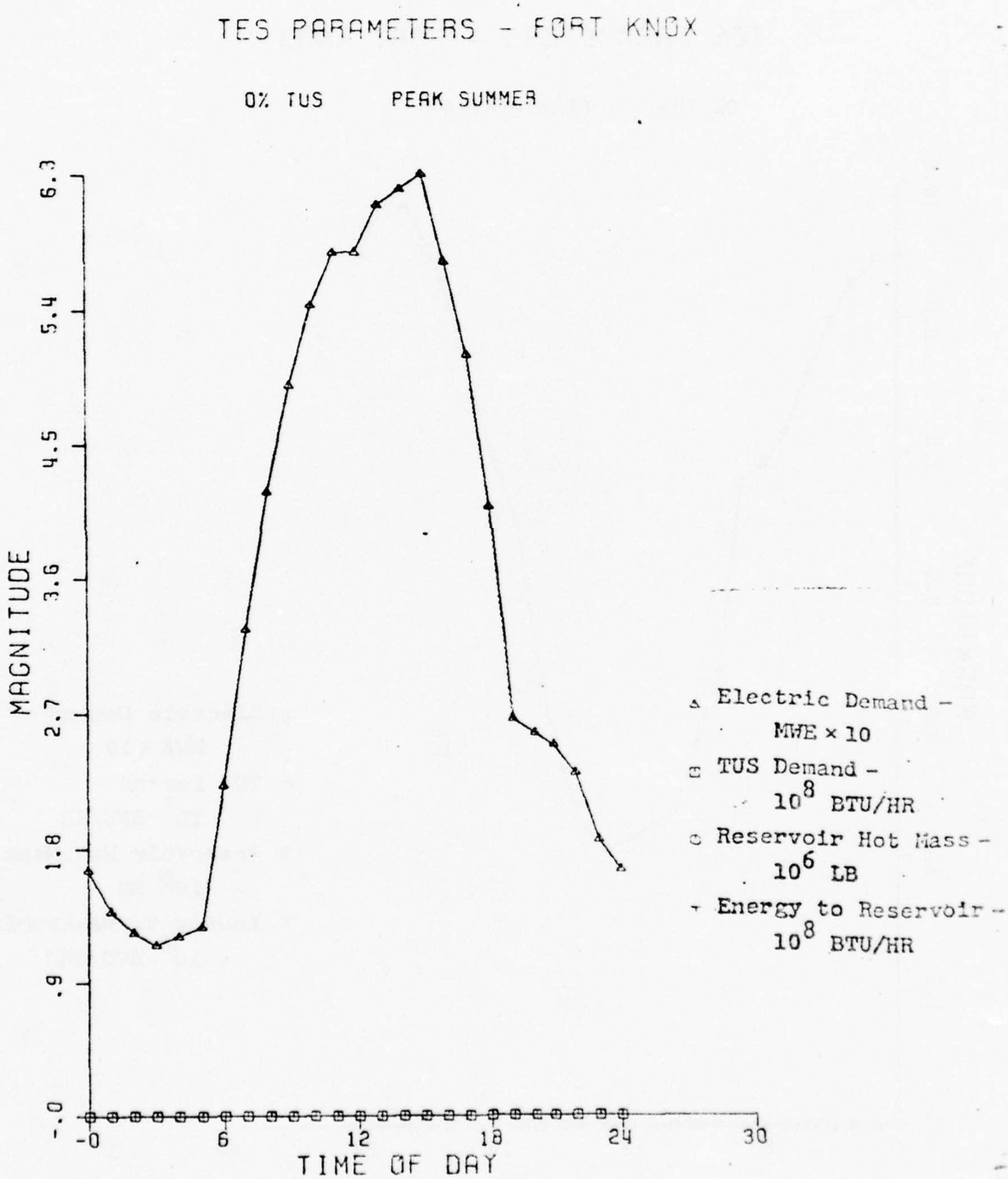

TES PARAMETERS - FORT KNOX
0% TUS PEAK SUMMER
MAGNITUDE
6.3
5.4
4.5
3.6
2.7
1.8
.9
-.0
-0
6
12
18
24
30
TIME OF DAY
Electric Demand - MWE × 10
TUS Demand - 10^8 BTU/HR
Reservoir Hot Mass - 10^6 LB
Energy to Reservoir - 10^8 BTU/HR

Figure 5.21

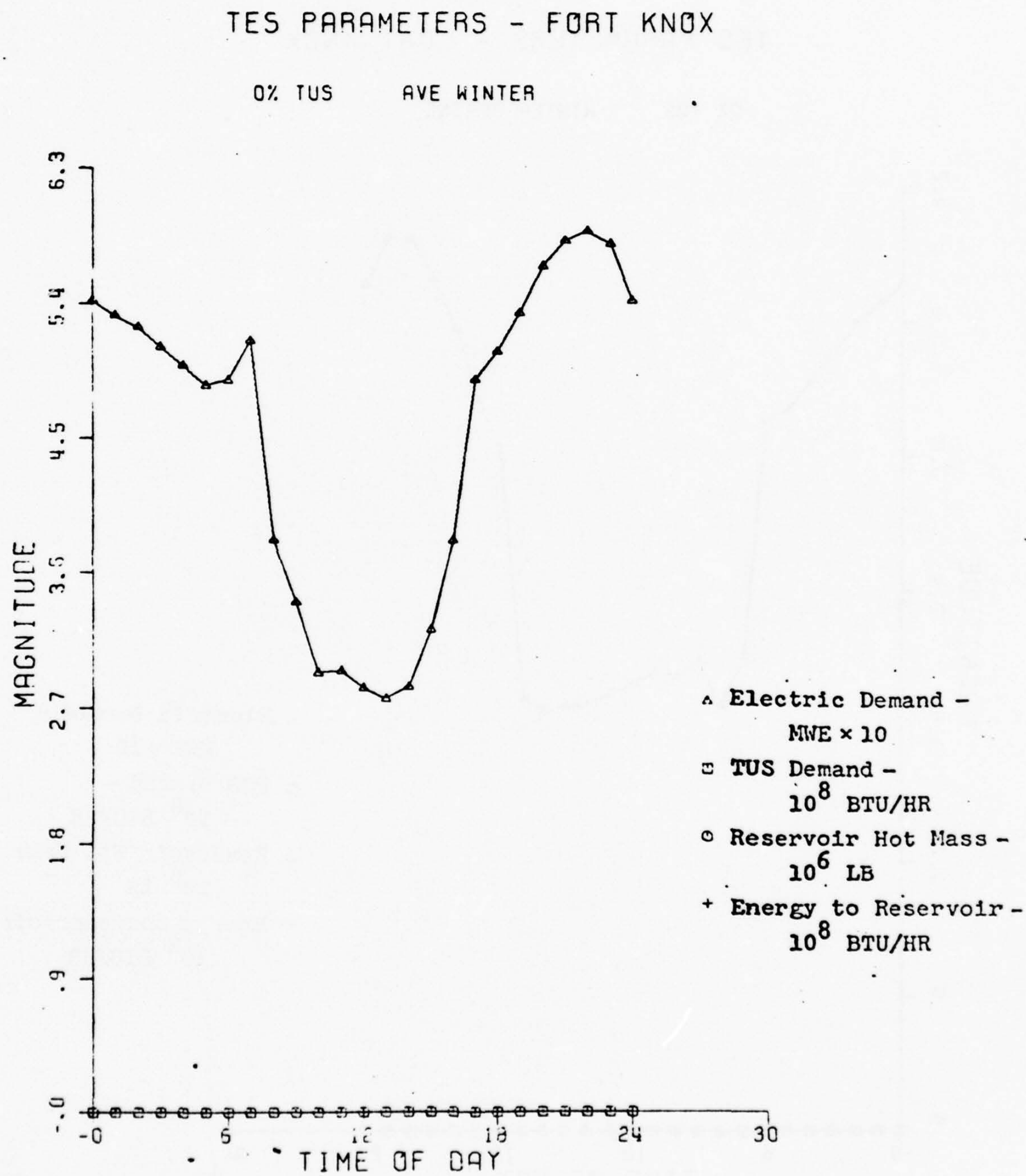

TES PARAMETERS - FORT KNOX
0% TUS AVE WINTER
MAGNITUDE
6.3
5.4
4.5
3.6
2.7
1.8
.9
-.0
-0
6
12
18
24
30
TIME OF DAY
Electric Demand - MWE × 10
TUS Demand - 10^8 BTU/HR
Reservoir Hot Mass - 10^6 LB
Energy to Reservoir - 10^8 BTU/HR

Figure 5.22

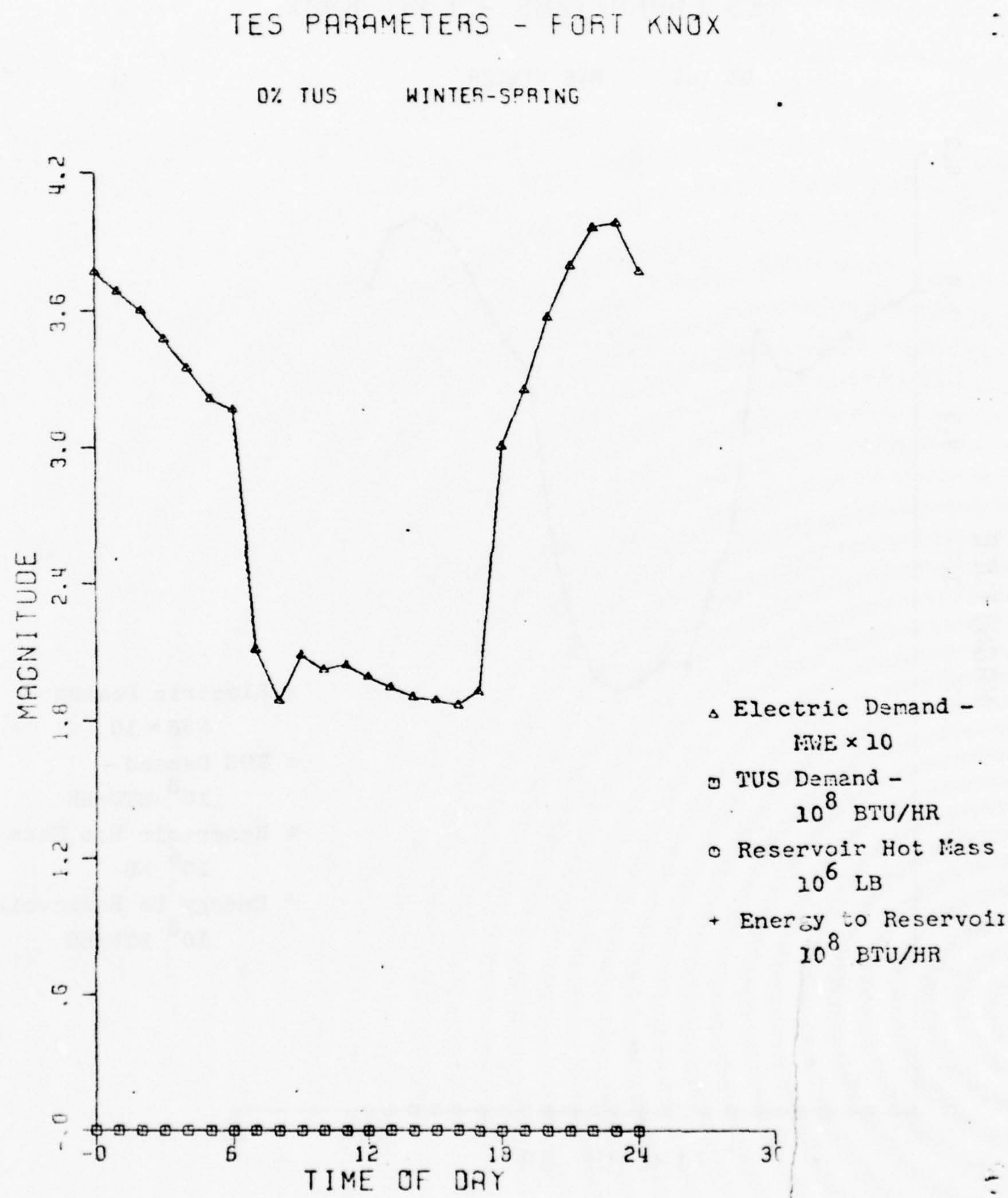

TES PARAMETERS - FORT KNOX
0% TUS WINTER-SPRING
MAGNITUDE
4.2
3.6
3.0
2.4
1.8
1.2
.6
-.0
-0
6
12
18
24
30
TIME OF DAY
Electric Demand - MWE × 10
TUS Demand - 10^8 BTU/HR
Reservoir Hot Mass 10^6 LB
Energy to Reservoir 10^8 BTU/HR

Figure 5.23

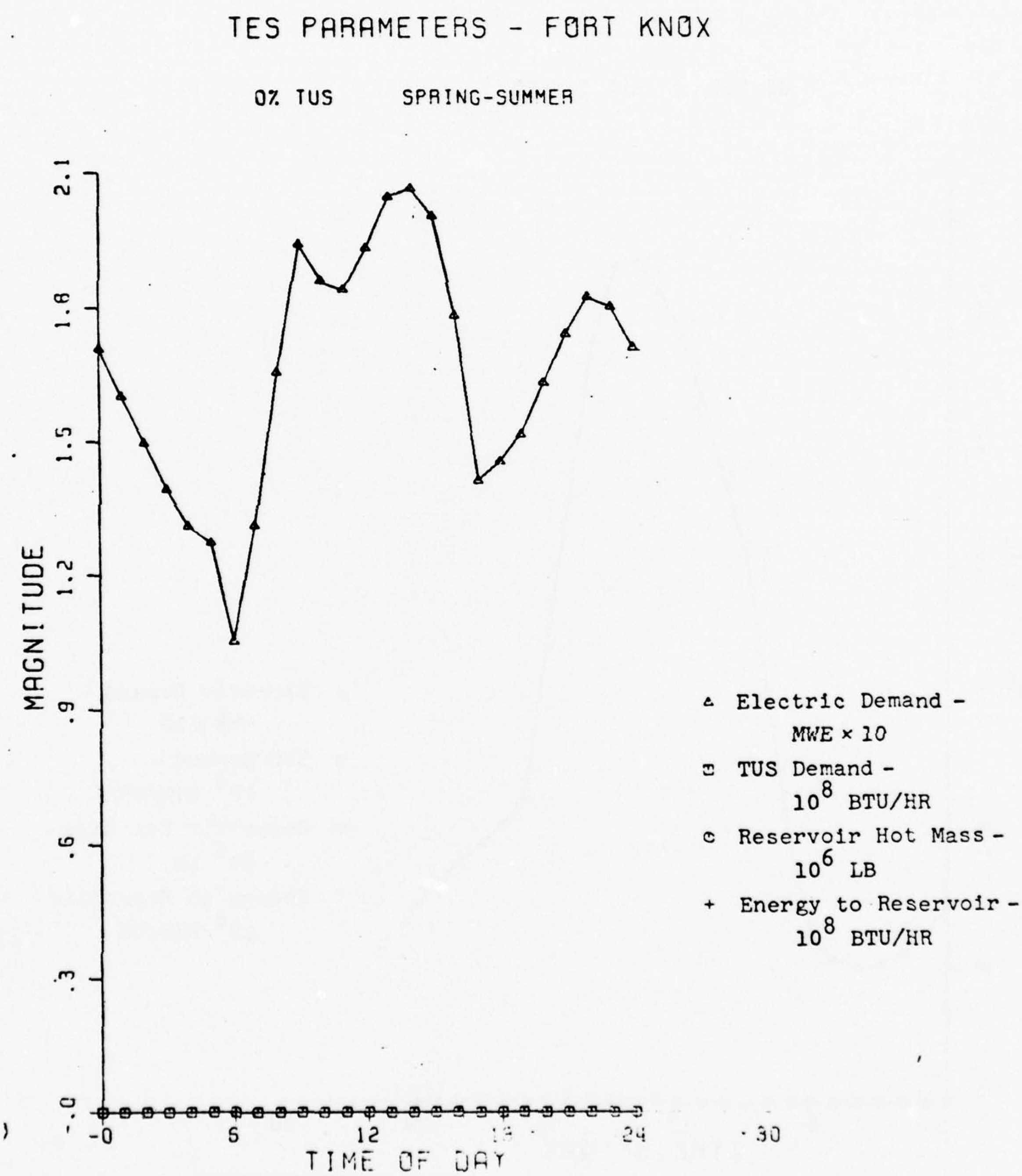

TES PARAMETERS - FØRT KNOX
0% TUS SPRING-SUMMER
MAGNITUDE
2.1
1.8
1.5
1.2
.9
.6
.3
-.0
-0
6
12
24
30
TIME OF DAY
Electric Demand - MWE × 10
TUS Demand - 10^8 BTU/HR
Reservoir Hot Mass - 10^6 LB
Energy to Reservoir - 10^8 BTU/HR

Figure 5.24

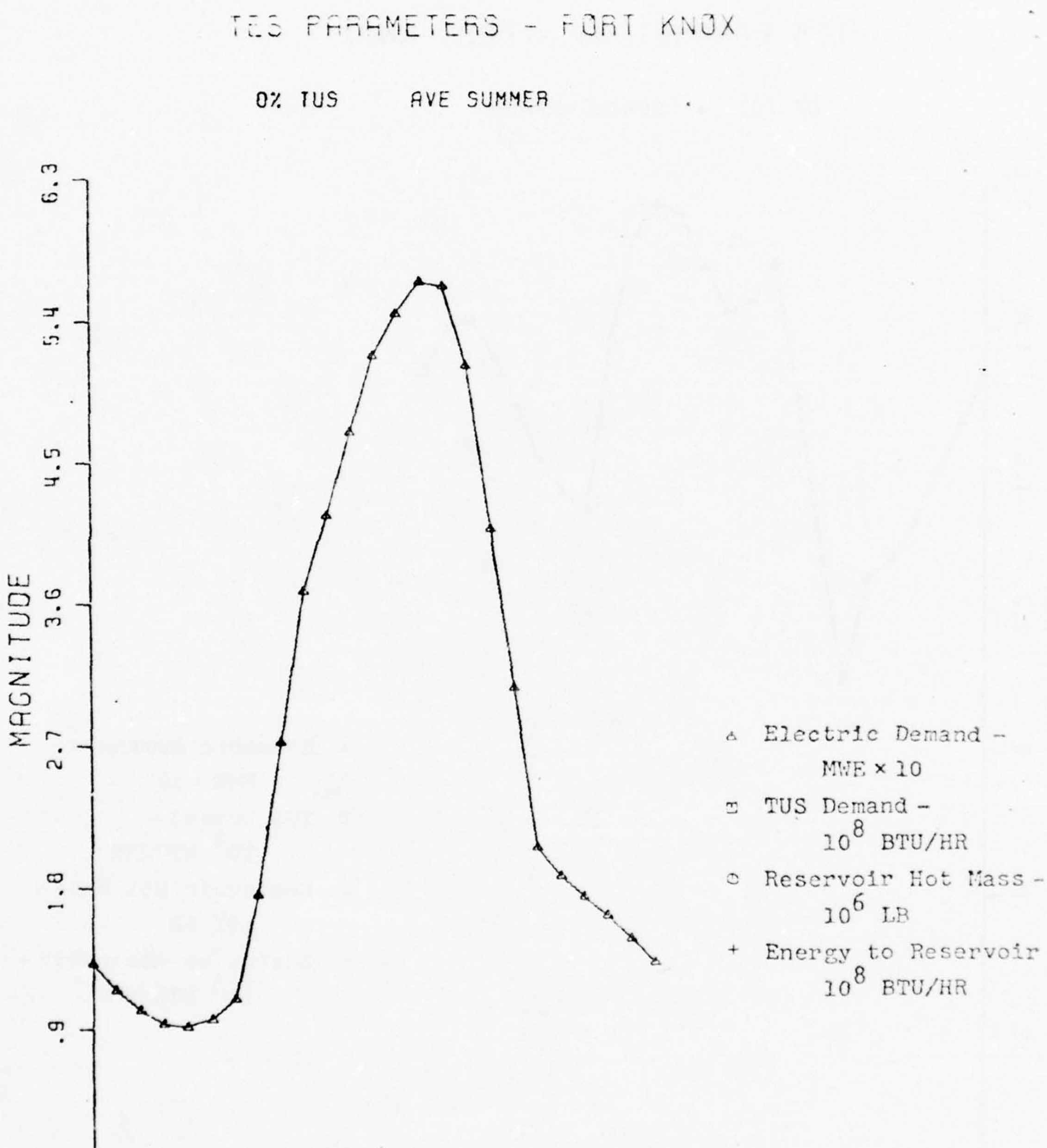

TES PARAMETERS - FORT KNOX
0% TUS AVE SUMMER
MAGNITUDE
6.3
5.4
4.5
3.6
2.7
1.8
.9
-.0
-0
6
12
18
24
30
TIME OF DAY
Electric Demand - MWE × 10
TUS Demand - 10^8 BTU/HR
Reservoir Hot Mass - 10^6 LB
Energy to Reservoir 10^8 BTU/HR

Figure 5.25

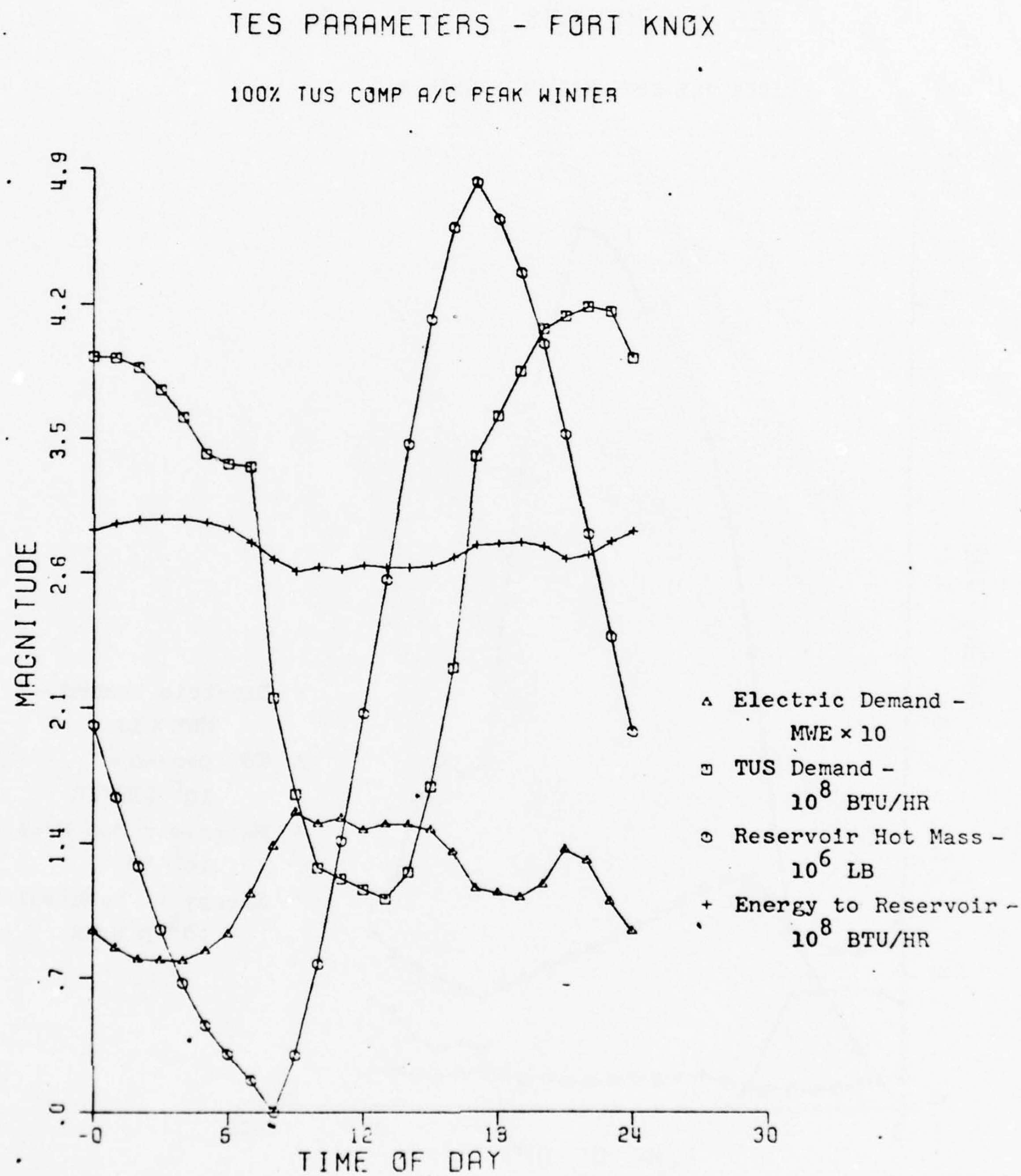

TES PARAMETERS - FORT KNOX
100% TUS COMP A/C PEAK WINTER
MAGNITUDE
4.9
4.2
3.5
2.6
2.1
1.4
.7
.0
-0
6
12
18
24
30
TIME OF DAY
Electric Demand - MWE × 10
TUS Demand - 10^8 BTU/HR
Reservoir Hot Mass - 10^6 LB
Energy to Reservoir - 10^8 BTU/HR

Figure 5.26

TES PARAMETERS - FORT KNOX

100% TUS COMP A/C PEAK SUMMER

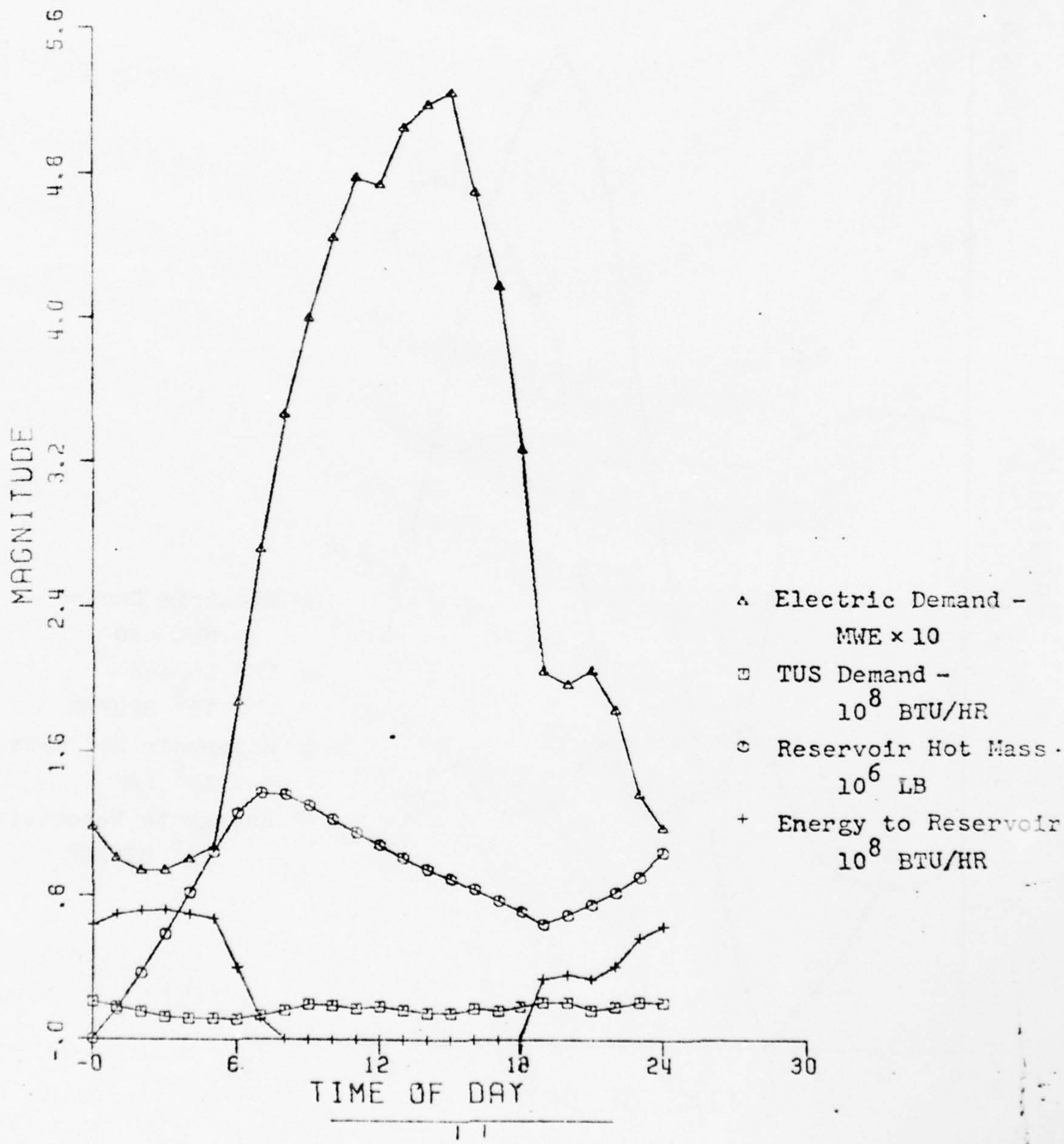


Figure 5.27

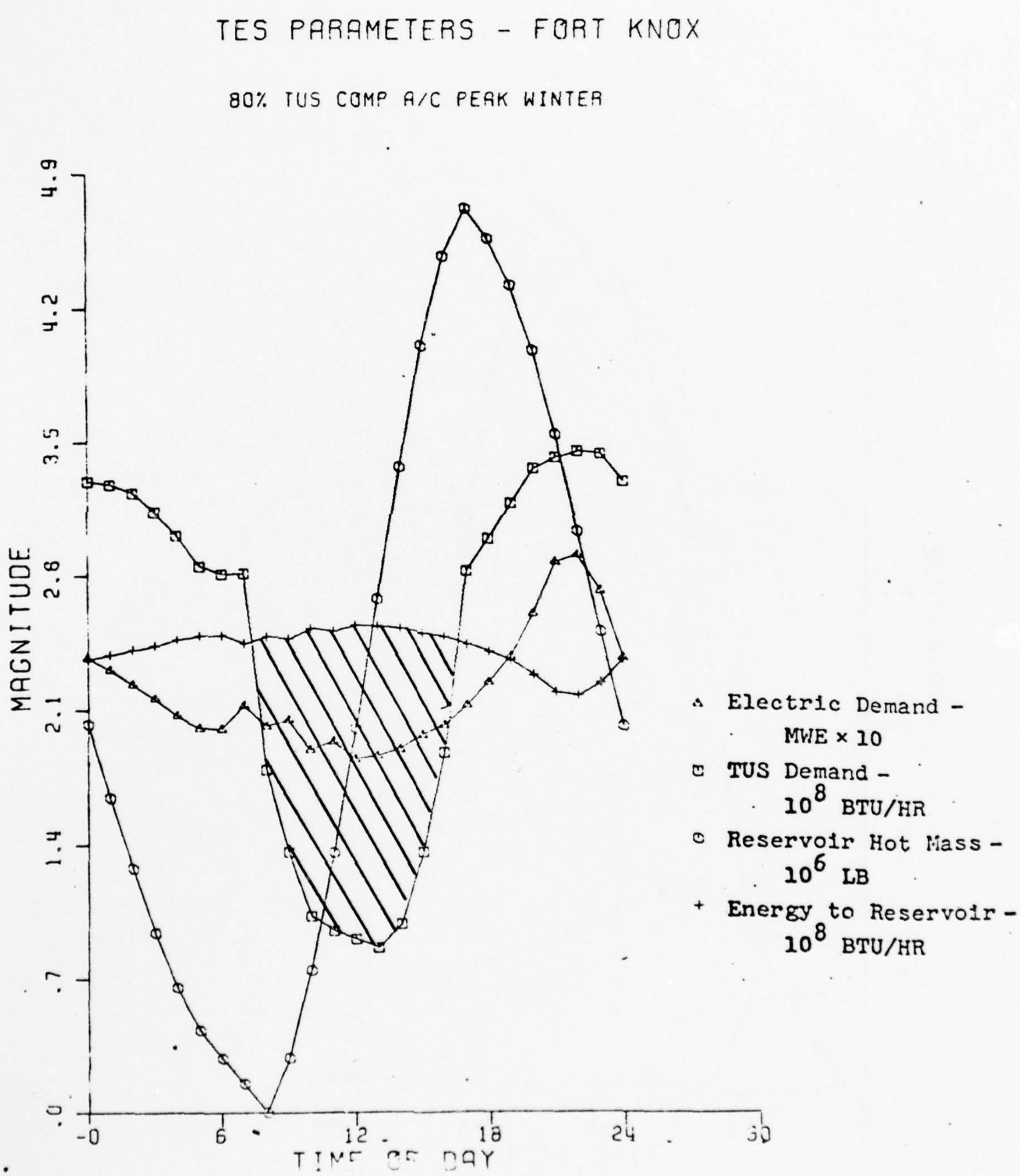

TES PARAMETERS - FORT KNOX
80% TUS COMP A/C PEAK WINTER
MAGNITUDE
.0
.7
1.4
2.1
2.8
3.5
4.2
4.9
-0
6
12
18
24
30
TIME OF DAY
Electric Demand - MWE × 10
TUS Demand - 10^8 BTU/HR
Reservoir Hot Mass - 10^6 LB
Energy to Reservoir - 10^8 BTU/HR

Figure 5.28

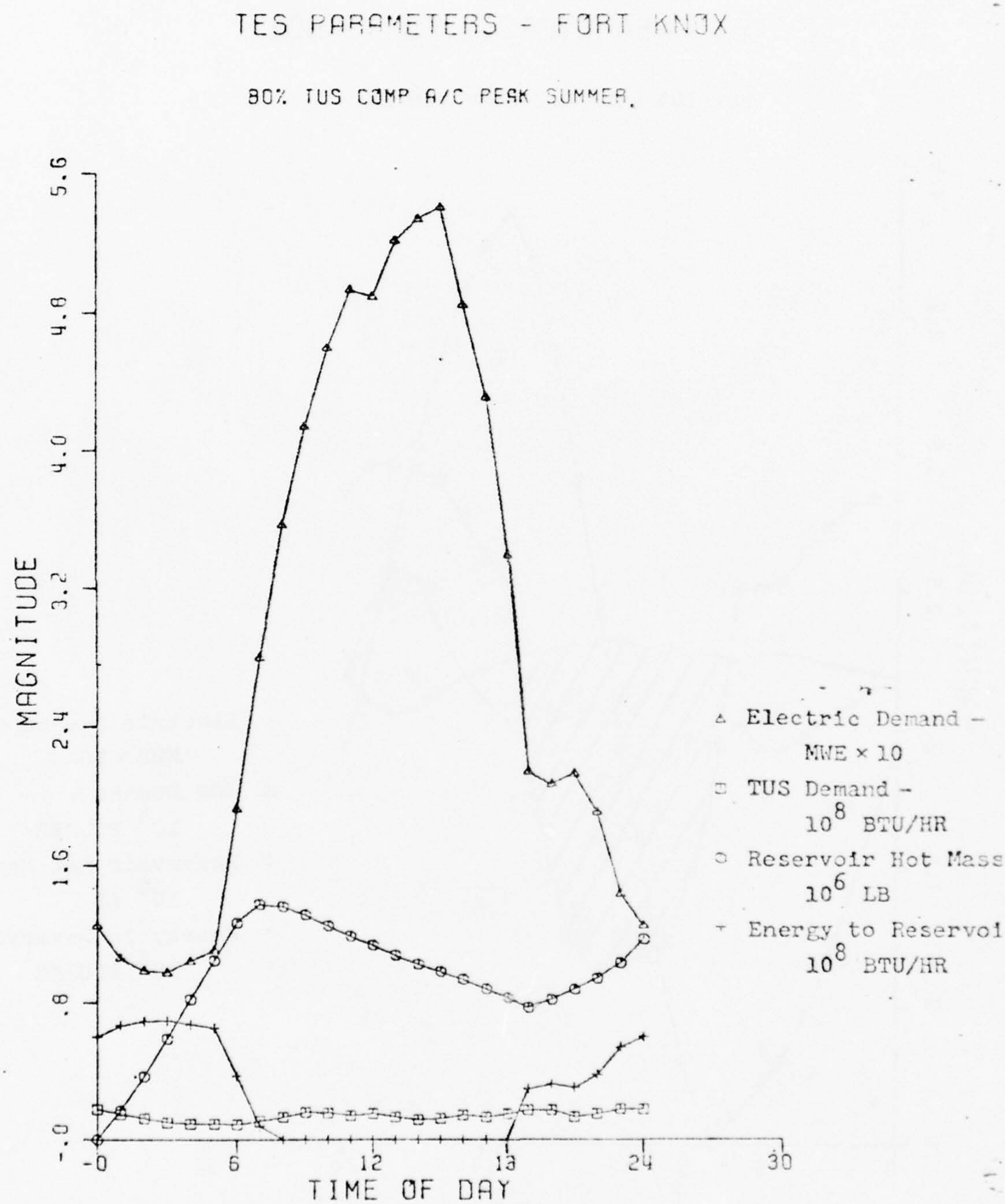

TES PARAMETERS - FORT KNOX
80% TUS COMP A/C PEAK SUMMER
MAGNITUDE
-.0
.8
1.6
2.4
3.2
4.0
4.8
5.6
-0
6
12
18
24
30
TIME OF DAY
Electric Demand - MWE × 10
TUS Demand - 10^8 BTU/HR
Reservoir Hot Mass 10^6 LB
Energy to Reservoi 10^8 BTU/HR

Figure 5.29

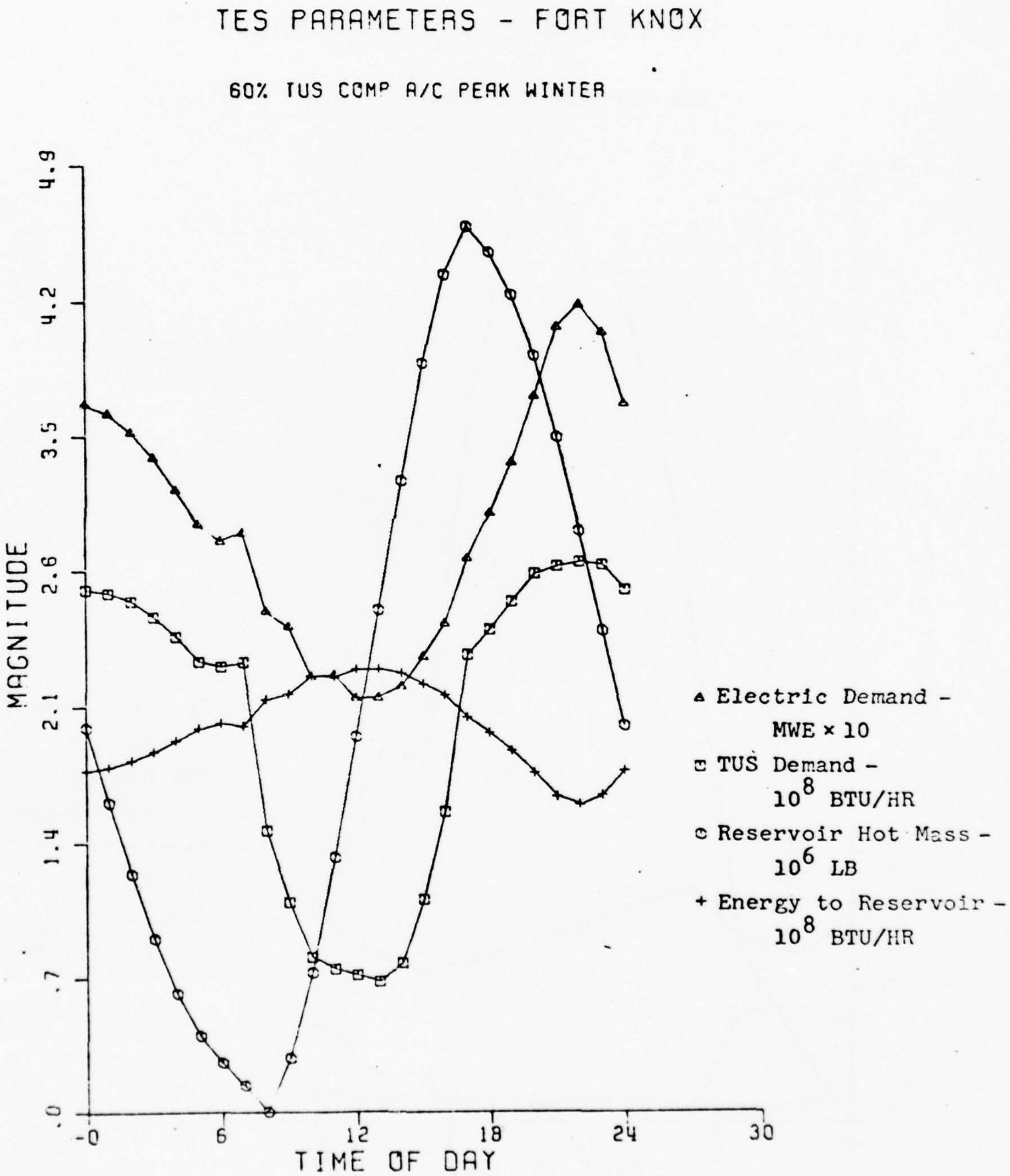

TES PARAMETERS - FORT KNOX
60% TUS COMP A/C PEAK WINTER
MAGNITUDE
4.9
4.2
3.5
2.6
2.1
1.4
.7
.0
-0
6
12
18
24
30
TIME OF DAY
Electric Demand - MWE × 10
TUS Demand - 10^8 BTU/HR
Reservoir Hot Mass - 10^6 LB
Energy to Reservoir - 10^8 BTU/HR

Figure 5.30

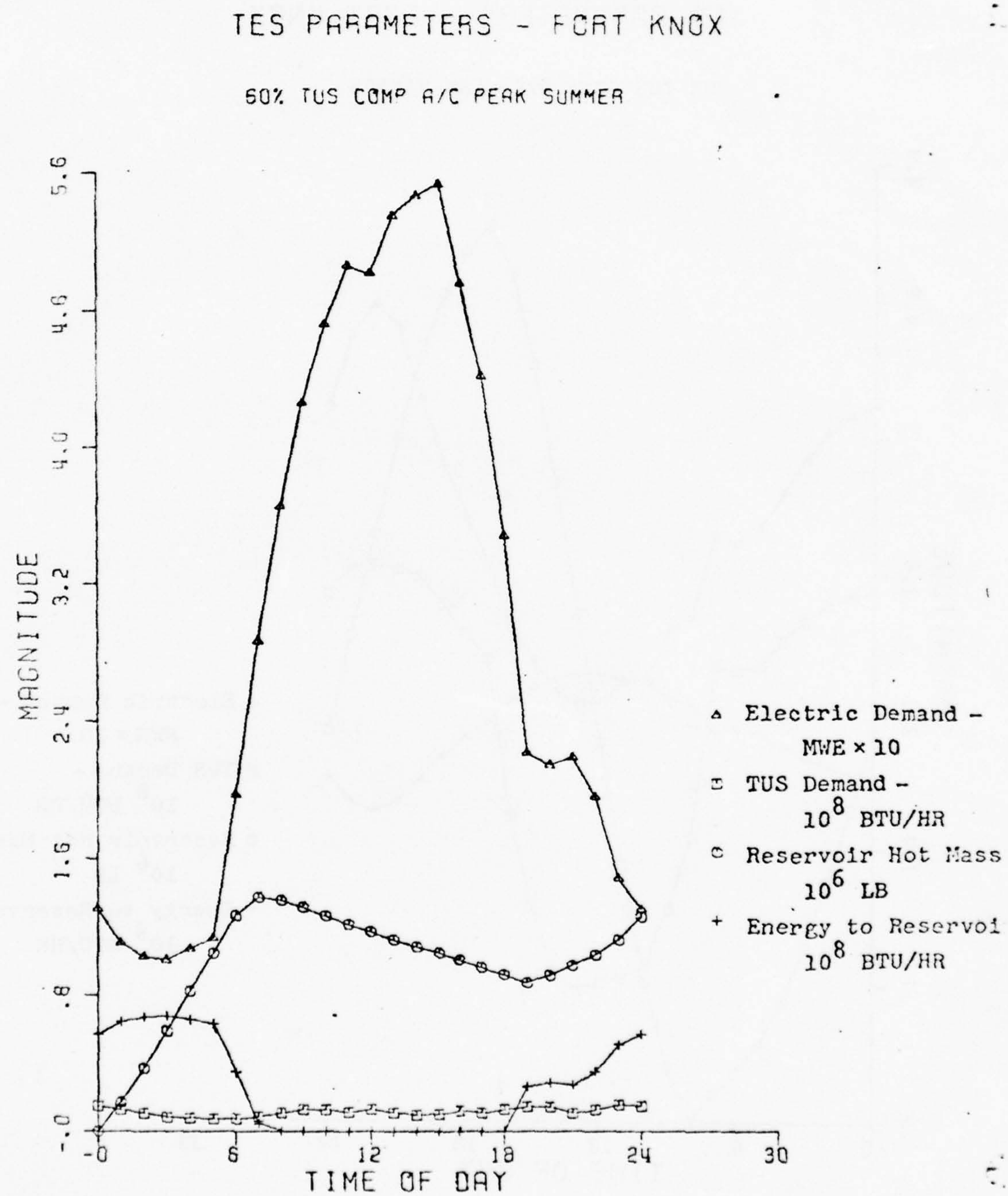

TES PARAMETERS - FORT KNOX
60% TUS COMP A/C PEAK SUMMER
MAGNITUDE
5.6
4.8
4.0
3.2
2.4
1.6
.8
-.0
-0
6
12
18
24
30
TIME OF DAY
Electric Demand - MWE × 10
TUS Demand - 10^8 BTU/HR
Reservoir Hot Mass 10^6 LB
Energy to Reservoi 10^8 BTU/HR

Figure 5.31

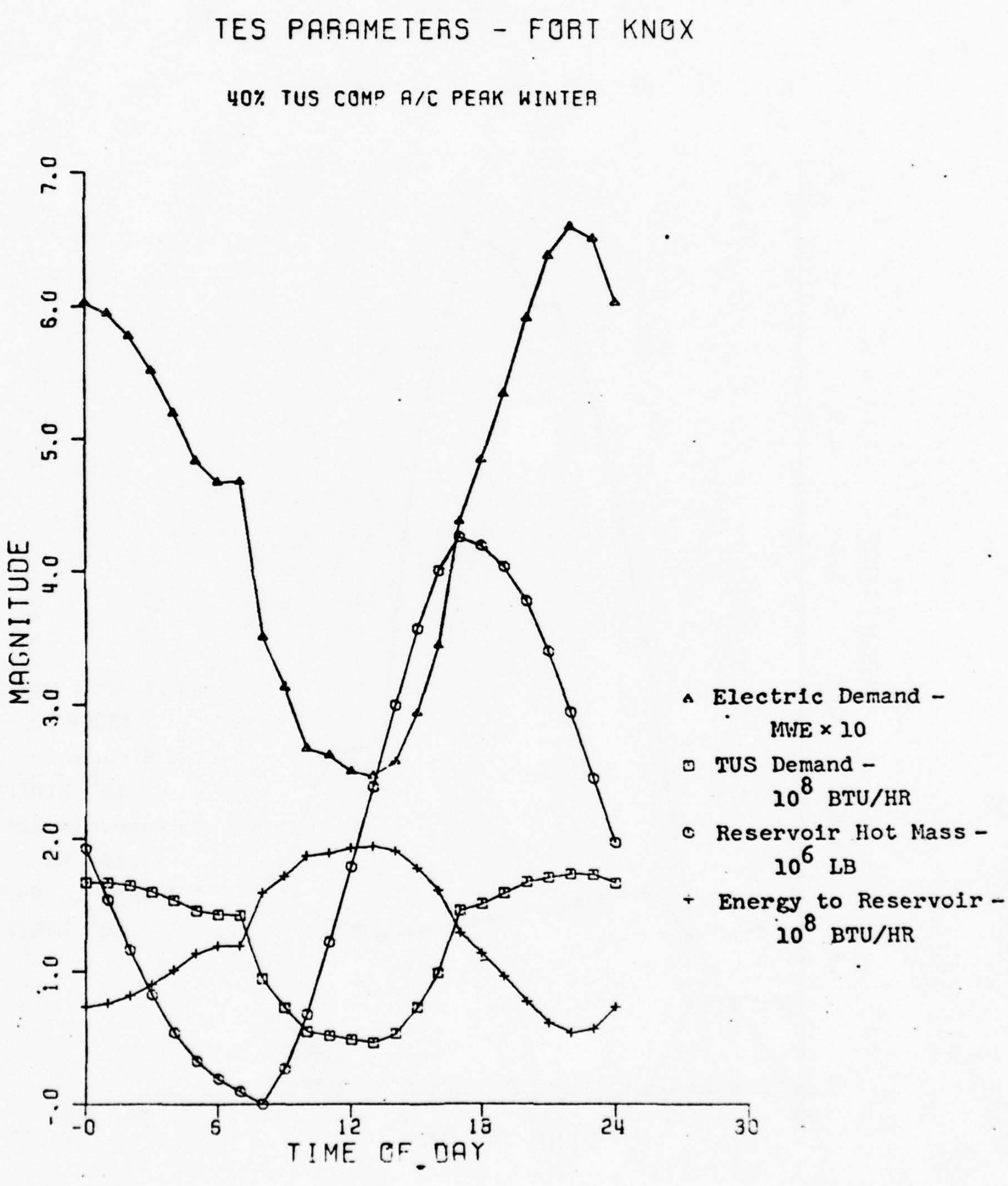

TES PARAMETERS - FORT KNOX
40% TUS COMP A/C PEAK WINTER
MAGNITUDE
7.0
6.0
5.0
4.0
3.0
2.0
1.0
-.0
-0
6
12
18
24
30
TIME OF DAY
Electric Demand - MWE × 10
TUS Demand - 10^8 BTU/HR
Reservoir Hot Mass - 10^6 LB
Energy to Reservoir - 10^8 BTU/HR

Figure 5.32

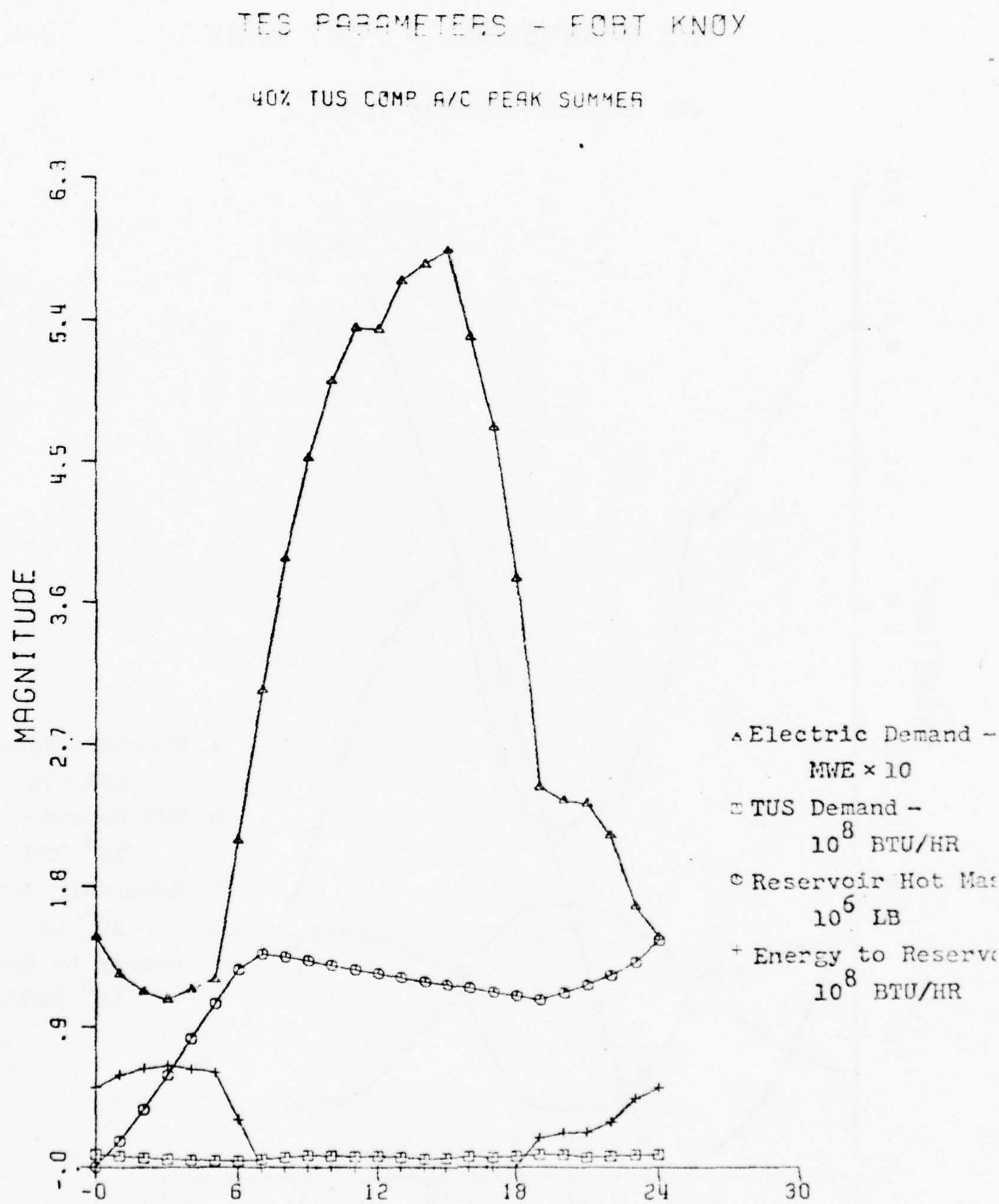

TES PARAMETERS - FORT KNOX
40% TUS COMP A/C PEAK SUMMER
MAGNITUDE
6.3
5.4
4.5
3.6
2.7
1.8
.9
-.0
-0
6
12
18
24
30
TIME OF DAY
Electric Demand - MWE × 10
TUS Demand - 10^8 BTU/HR
Reservoir Hot Mas 10^6 LB
Energy to Reserv 10^8 BTU/HR

Figure 5.33

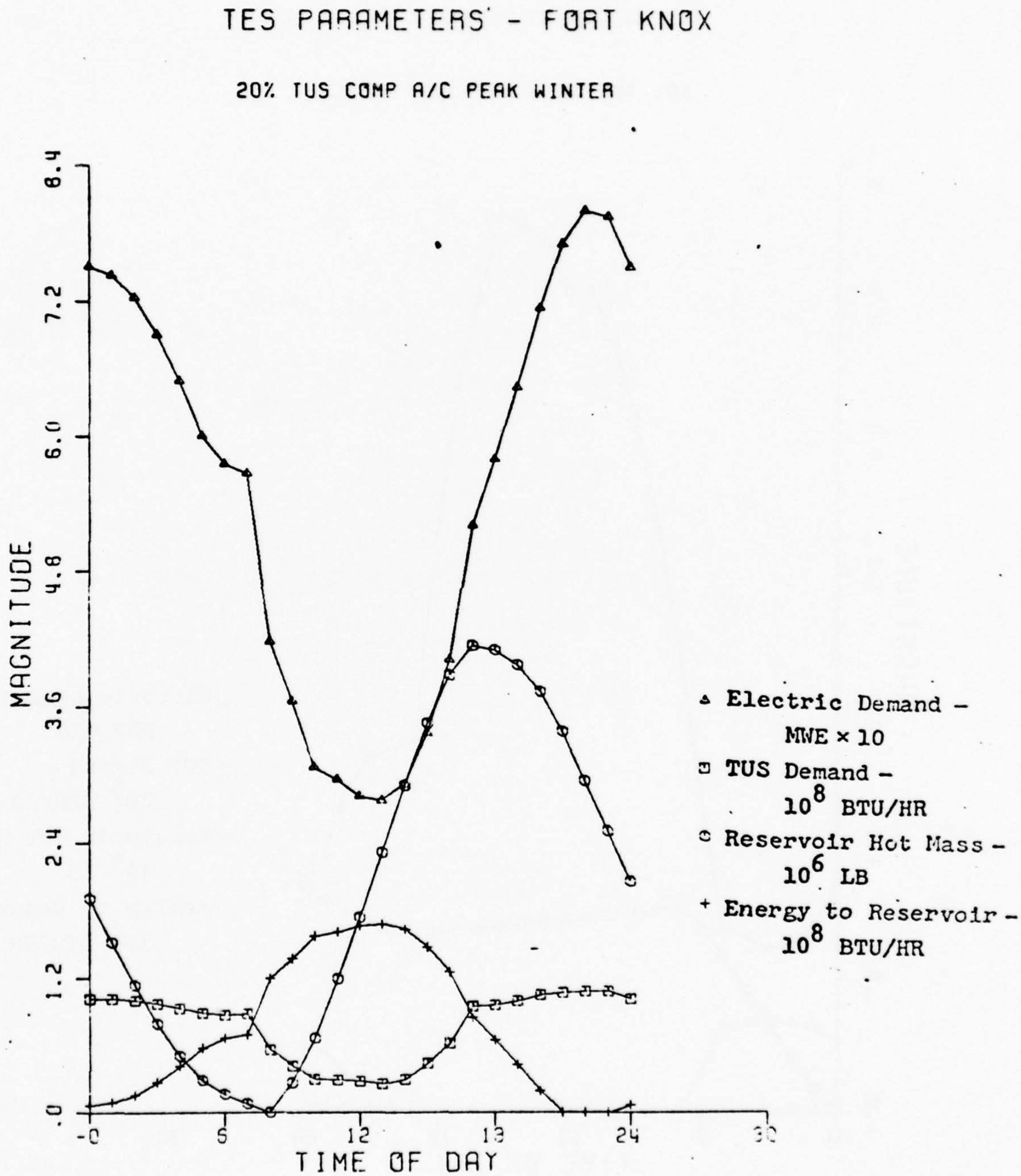

TES PARAMETERS - FORT KNOX
20% TUS COMP A/C PEAK WINTER
MAGNITUDE
.0
1.2
2.4
3.6
4.8
6.0
7.2
8.4
-0
6
12
18
24
30
TIME OF DAY
Electric Demand - MWE × 10
TUS Demand - 10^8 BTU/HR
Reservoir Hot Mass - 10^6 LB
Energy to Reservoir - 10^8 BTU/HR

Figure 5.34

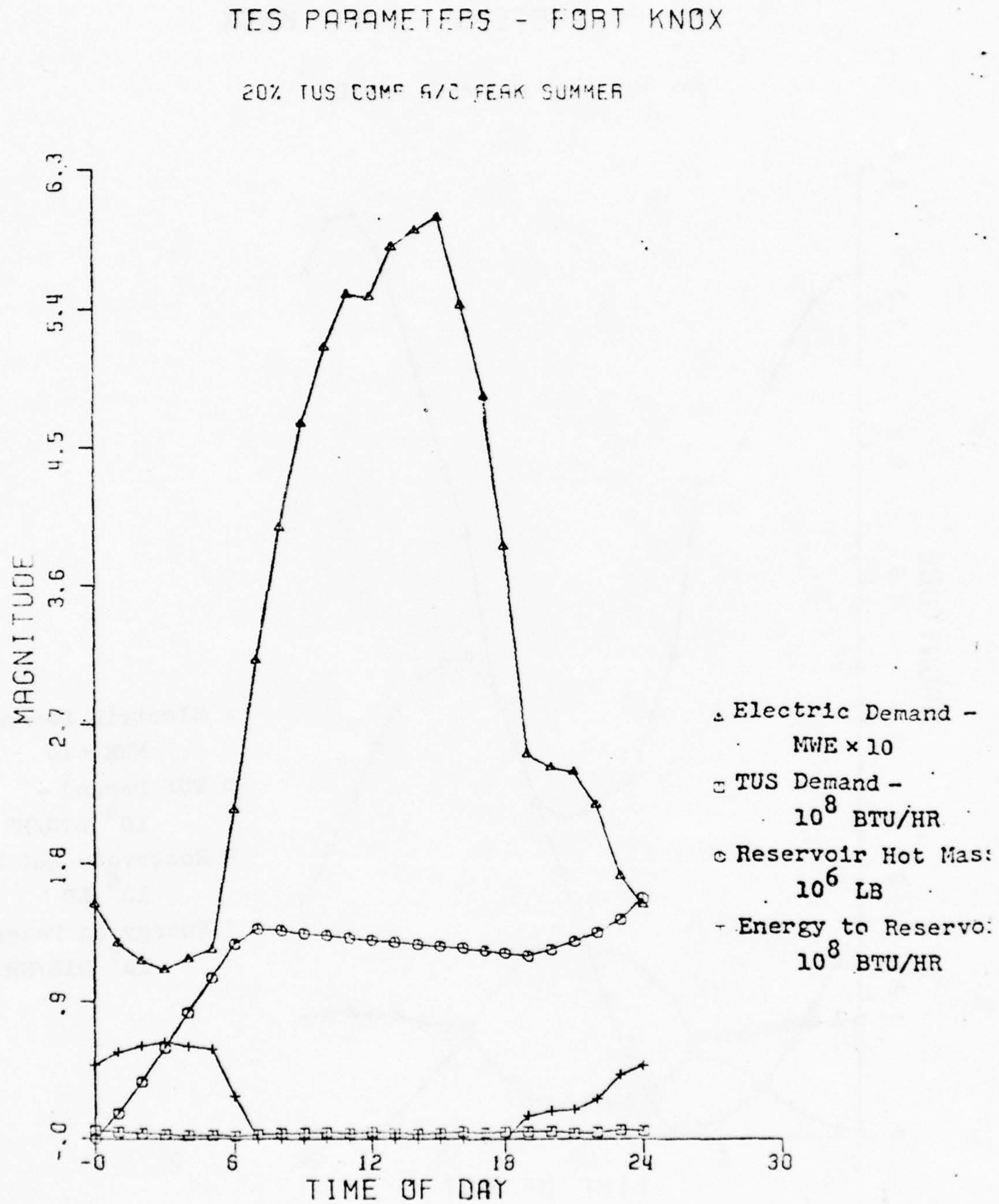

TES PARAMETERS - FORT KNOX
20% TUS COMP A/C PEAK SUMMER
MAGNITUDE
6.3
5.4
4.5
3.6
2.7
1.8
.9
-.0
-0
6
12
18
24
30
TIME OF DAY
Electric Demand - MWE × 10
TUS Demand - 10^8 BTU/HR
Reservoir Hot Mas: 10^6 LB
Energy to Reservo: 10^8 BTU/HR

Table 5.1 lists the required reactor size (in MW(t)) and reservoir volume for each split value for each of the two air conditioning options for winter peak and summer peak weather conditions. The asterisk indicates the size required to meet the annual system power demand. Thus, for the 100% split value of option 1 (use of absorptive air conditioning), the required reactor size is the 97.1 MW(t) required for the peak winter day, not the 76.4 MW(t) required for the peak summer day. Similarly the required reservoir volume is the 11.4×10^4 ft^3 required for the peak summer day, not the 8.9×10^4 ft^3 required for the peak winter day.

Trends in component sizing conform to expectations. For Option 1, as the thermal/electrical split value is reduced from 100%, the required reactor size first decreases to a minimum, and then increases. At high split values insufficient waste heat is produced from electrical energy generation, and additional thermal power capacity is required for direct TUS service. The initial decrease in required reactor size is due to the effect of electrically powered heat pumps replacing TUS heating. The heat pumps have a nominal COP of approximately 2.2, so that in decreasing to a split value of approximately 80%, the net energy supplied by the core decreases although the gross end-use heating demand remains unchanged. However, as the split value is further reduced, the increasing amount of

TABLE 5.1

PLANT SIZE (MW(t))/RESERVOIR SIZE (Ft^3)

Split	Season of Design Day	Absorptive Air Conditioning-Electric Hot Water Option (1)		Compressive Air Conditioning-TUS Water Heating Option (2)	
		Power Plant Size (MW(t))	Thermal Reservoir Size (Ft^3)	Power Plant Size (MW(t))	Thermal Reservoir Size (Ft^3)
100%	W	97.1*	8.9×10^4	95.9	$8.9\times10^{4*}$
	S	76.4	$11.2\times10^{4*}$	138.0*	NA
80%	W	93.4*	8.6×10^4	92.4	$8.6\times10^{4*}$
	S	70.65	$10.3\times10^{4*}$	142.3*	NA
60%	W	116.2*	8.4×10^4	110.1	$8.4\times10^{4*}$
	S	73.4	$9.5\times10^{4*}$	145.6*	NA
40%	W	176.6*	$7.8\times10^{4*}$	173.6*	$7.8\times10^{4*}$
	S	100.4	7.6×10^4	153.7	NA
20%	W	212.9*	$7.6\times10^{4*}$	210.7*	$7.6\times10^{4*}$
	S	114.0	6.7×10^4	157.6	NA
0%	W	270.3*	NA	SAME	NA
	S	165.8			

* indicates required component size for satisfactory year-round operation.

electrical power required to drive the heat pumps results in power plant thermal energy production output being greater than the thermal demand. This requires an increased reactor size and results in wasted thermal energy. It is only at the unique split value of approximately 80% that the waste heat from electrical production matches the day-long TUS thermal energy demand, and results in a minimum core size being required. This trend of increasing production of non-usable waste heat production continues as the split value decreases, until at a split value of 0% all of the waste heat from electrical generation must be dissipated fruitlessly.

Required thermal reservoir size for power summer peaks in Option 2 are not shown, because the only thermal demand consists of the small domestic hot water demand. This thermal power demand is always instantaneously less than the waste heat power available, and therefore no reservoir is required.

As expected, the required reactor sizes for summer peak loads of Option 2 (compressive air conditioning) are larger than the reactor sizes of Option 1 (absorptive air conditioning). In essence, during the summer Option 2 uses only the plant's electrical output, and the waste heat from electrical generation goes unused. However, the absorptive air conditioning of Option 1 utilizes this waste heat and therefore this option requires a smaller power plant.

Similarly, the 0% winter case (i.e., all heat pump heating) uses only the plant's electrical power output, and wastes all of the thermal energy. Note, however, that for the 0% case the required reactor size is based on winter heating loads since these are larger than summer cooling loads.

5.2 Sizing of the Thermal Energy Storage Reservoir

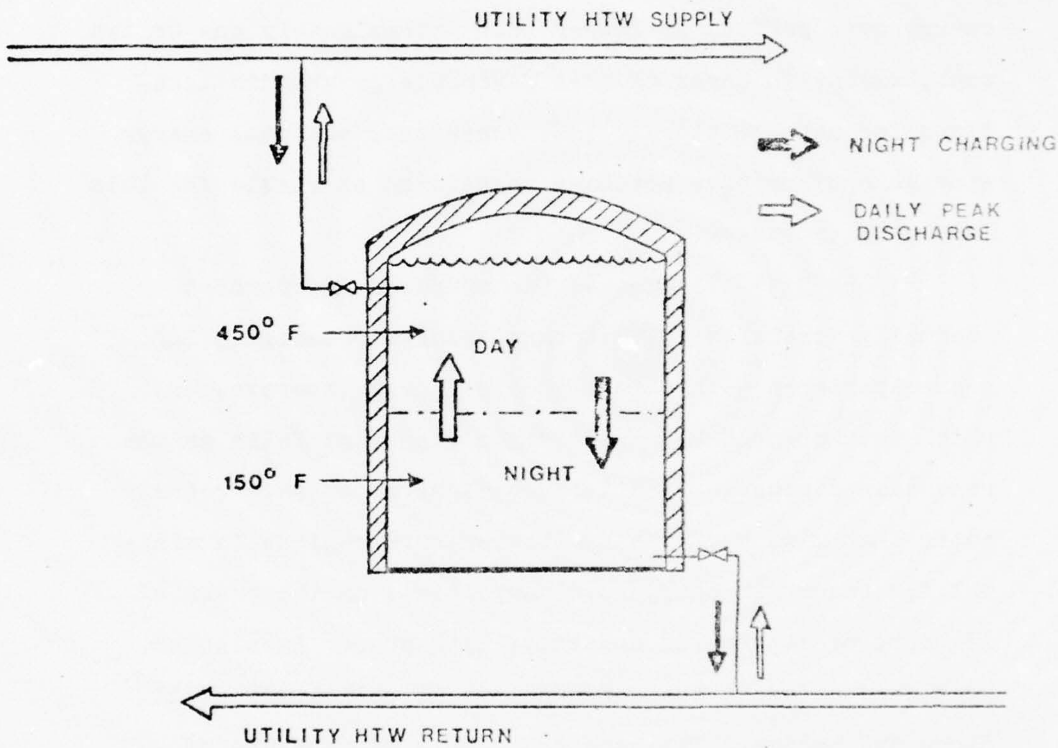
The primary function of the thermal energy storage reservoir is to supply the thermal utility system power demands during periods of insufficient power plant output. Its size therefore depends critically upon the assumed mode of power plant operation and upon the thermal energy supply and demand imbalances determined by the variations in the thermal and electrical load schedules. In the preceding section a constant total energy output mode of plant operation has been described, in which the station's electrical power output follows its electrical demand schedule, and its thermal output is buffered from the thermal demand variations by the reservoir. In general, because Fort Knox's thermal and electrical energy demand peaks are not of a comparable magnitude for all non-optimal conditions (e.g., a 38% efficient power plant cannot generally produce electricity to meet the peak electrical demand and use directly its turbine exhaust heat to match the thermal peak) and because these peaks occur at different times during the day (see Section 5.1), the reservoir must be sized

to store a relatively large quantity of hot water for periods of 12 hours or more. (Present technology precludes the efficient storage and retrieval of large amounts of energy over periods of longer than approximately one or two days, except in cases of most favorable geographic locations for natural storage [1]; therefore, seasonal energy storage options have not been considered as viable for this 1985 utility system.)

Figure 5.35 illustrates the proposed underground thermally stratified HTW storage reservoir design. European experience with this method of energy storage, in which the hot/cold water interface rises and falls as the reservoir discharges and is charged with hot water, indicates that mixing of the two temperature regions is minimal for reservoir charge/discharge times on the order of 12 hours or longer [1] and that, with proper insulation, heat conduction to the surroundings is relatively small. Since the proposed TES uses water at a temperature of 380°F as the primary supply to the thermal utility system, the reservoir must be pressurized to roughly 250 psia to prevent the HTW's flashing to steam. To date, cylindrical steel storage tanks capable of withstanding this pressure have been limited to a size of roughly 20 feet in diameter and 70 feet long [2]. From Section 5.1, it is seen that the 21,991 cubic-foot volume of one of these tanks does not provide enough capacity to smooth the thermal

Figure 5.35

Underground Stratified Thermal Energy Storage Reservoir



energy supply and demand imbalances. Therefore, the proposed storage reservoir is not to be a single cylindrical tank as indicated by the dimensions referenced in Section 5.1, but it is rather composed of a set of these smaller tanks piped in series to provide the required storage volume and thermal stratification.

Although the daily simulation descriptions in Section 5.1 present reservoir sizing data in the context of systems designed individually for each of the six days studied, only one day out of the year actually governs the size of the reservoir to be installed for each of the utility system options. Since the reservoir volume is determined by the maximum discrepancy between the thermal energy supply and demand schedules and not by the absolute magnitudes of these energy flows, the primary criterion to be met by the storage system is that on the most severely imbalanced day of the year, the reservoir, in combination with the thermal energy output of the power plant, must supply enough thermal energy to just meet the utility system demands without being completely discharged and must be fully recharged during the 12-hour period following the maximum mismatch (to be prepared for the next day's cycle). From Section 5.1 it is seen that for Option 1, although the day with the maximum total energy consumption is the peak winter design day, the peak summer day's thermal demand schedules exhibit the greatest variations, and therefore the energy supply and

and demand conditions on that day determine the reservoir size required for this option. Table 5.1 presents the reservoirs chosen for each of the utility system options as dictated by these requirements. If the reservoirs are sized to just meet the design volume criterion shown, they will be completely discharged only during the peak imbalance day, and the position of the hot/cold water interface will vary much less on the remainder of the days throughout the year.

5.3 Annual Energy Consumption

Using the thermal and electrical energy demands computed in each of the daily simulations, the annual energy consumption for a given case is calculated. Table 5.2 lists these data. The parameter shown in Table 5.2 is the total core thermal energy produced during the year for all of the cases examined. This is not equal to the power delivered to the end-use consumers, because throughout most of the year the thermal demand is less than the waste heat available from electrical generation, and this excess heat is exhausted to the atmosphere. Coal consumption rates are discussed in Section 6.

Table 5.2 shows that, of all the cases considered, maximum uranium conservation is achieved by the 80% split configuration of Option 1. Option 1 fuel use is less than Option 2 fuel use due to the constant power requirement

TABLE 5.2

ANNUAL ENERGY CONSUMPTION RATES* FOR

THE TES CONFIGURATIONS WHICH WERE EXAMINED

Split Value	Option 1 Absorptive Air Conditioning (Units of 10^5 MW(t)hr)	Option 2 Compressive Air Conditioning (Units of 10^5 MW(t)hr)
100%	5.34	6.50
80%	5.18	6.46
60%	5.71	6.81
40%	7.24	8.13
20%	8.13	8.92
0%	10.13	10.13 (same)

*MW(t)hr of core power generated annually.

imposed on plant operation. That is, the high core powers (required to supply peak electrical demands for compressive air conditioning in the summer) are not efficiently utilized. Therefore, even though the compressive air conditioning COP has a value of 2.2 compared to the absorptive air conditioning COP value of .95, the total core power required is larger for Option 2, because unused waste heat is produced during much of the day. If load following operation were used in Option 2, total energy consumption would decrease (see Section 6 - Coal Plant Operation). However, plant size would still be fixed by peak power demands and therefore would remain unchanged.

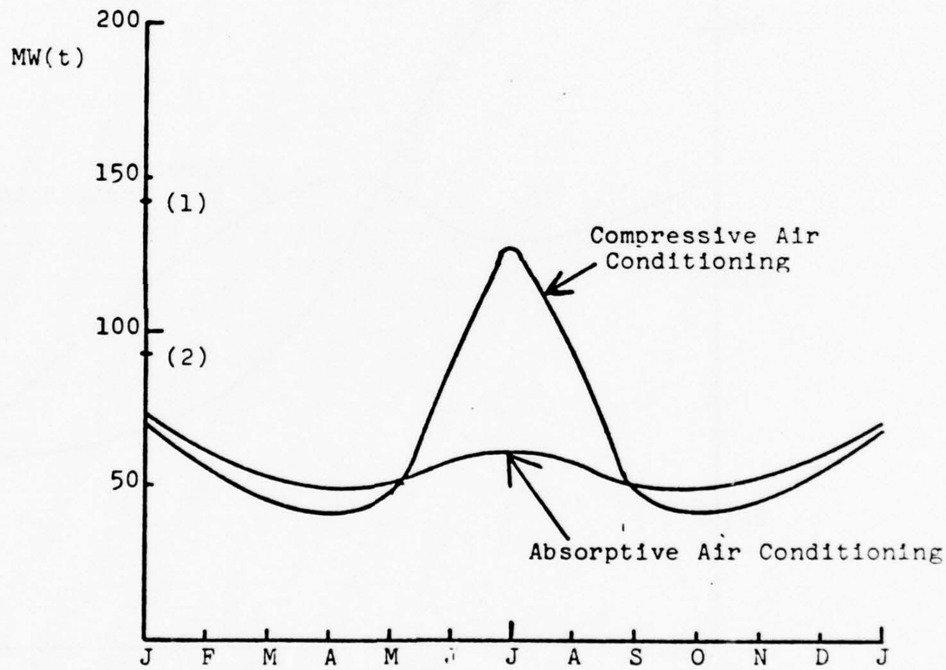
To further illustrate the differences between Options 1 and 2, Figure 5.36 shows a plot of the daily core power for both options as a function of the month of the year for an 80% split value. In generating these annual power schedules, it is assumed that the base's energy consumption characteristics are symmetrical between the spring and fall seasons.

Figure 5.36 shows clearly the unfavorable effect on constant power systems of using compressive air conditioning. Not only is the average daily power level higher during the summer for Option 2 than for Option 1, the plant capacity factors for Option 2 are lower than those of Option 1 (for high thermal/electric split values) as shown in Figure 5.37. Option 2 capacity factors for split values lower than 60%

FIGURE 5.36

FORT KNOX ANNUAL DAILY-AVERAGE

ENERGY CONSUMPTION SCHEDULE



Notes:

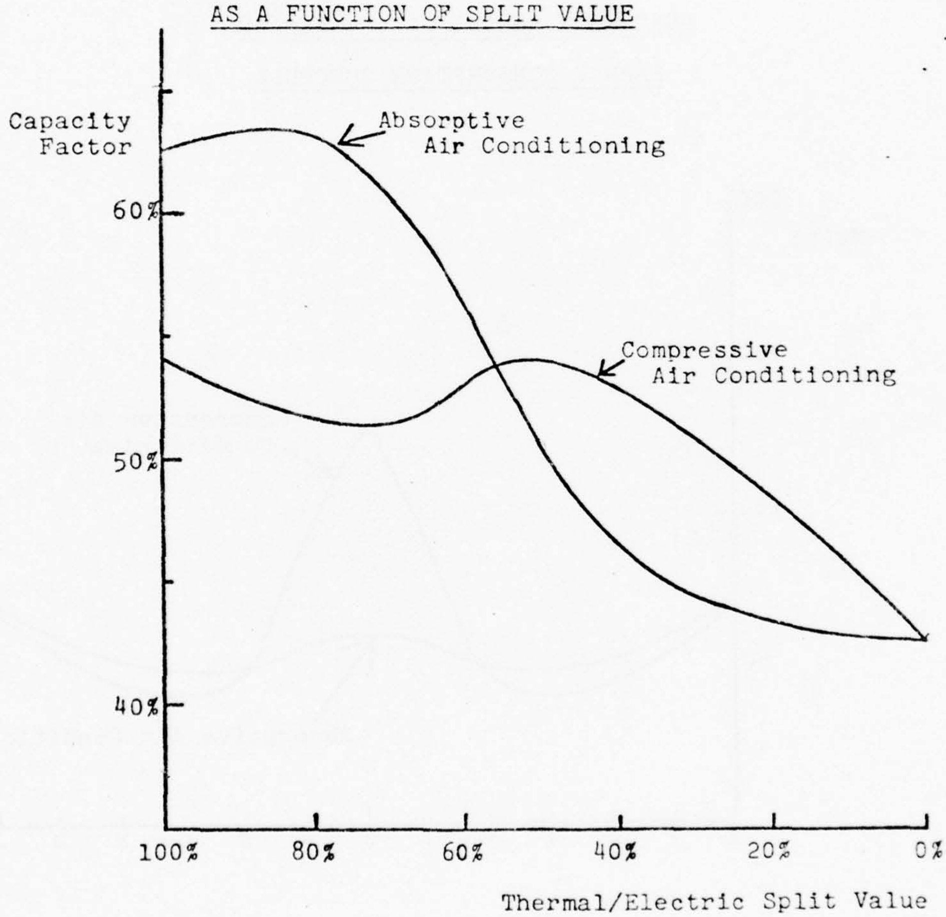
(1) Summer peak-sized core power = 142.3 MW(t), Option 2

(2) Winter peak-sized core power = 93.4 MW(t), Option 1

Figure 5.37

FORT KNOX PLANT CAPACITY FACTOR

AS A FUNCTION OF SPLIT VALUE



improve due to the increasing winter season electrical demand. At split values above 40%, Option 2 winter electrical demand is small and therefore the peak summer sized reactor operates at low power levels. For Option 2, as the split value falls below 60%, better use is being made of the reactor during the winter than is the case at higher split values for which winter electrical demand is low.

The argument can be made that if the plants were allowed to operate in a load-following manner, the excess energy production could be significantly reduced, if not eliminated entirely. However, because of the general desirability of operating a power plant at approximately constant total energy output levels and because operational variations do not affect the required installed capacity of the plant, the load-following operational characteristics for the nuclear option have not been studied in depth. It seems, however, that because of the relatively large electrical demands and relatively small thermal demands for most of the year, operating the power plants in a load-following mode would, at best, reduce the excess energy generation but would not completely eliminate it. Section 6 discusses the characteristics of the fossil-fired central station option operated in a load following mode.

REFERENCES

1. Feher, R.D., "Energy Storage Alternatives for a Nuclear Total Utility System," MIT Department of Nuclear Engineering Special Study Report (May 1974).

2. "An Assessment of Energy Storage Systems Suitable for Use by Electric Utilities," Volume 2, Public Service Electric and Gas Company, Newark, EPRI Research Project 225, (Dec. 1975).

CHAPTER 6

ECONOMIC OPTIMIZATION

6.1 Introduction

The economic optimization of the TES is discussed in five parts: the economics of the nuclear power plant as developed by Metcalfe et al. [1], the economics of the CGGT plant, the economics of the TUS, the economics of the electrical transmission distribution and end-use equipment, and the economics of the combined Total Energy System (TES). The standard of economic comparison used in this report is the cost of the proposed system in 1985 dollars. However, the data base and escalation rates used to project current costs to 1985 costs are also presented.

6.2 Nuclear Power Plant Costs

The size of the nuclear power plant required to supply the Fort Knox TES is presented in Section 5.1. Calculation of the component costs of the plant is detailed in Appendix D.1. The cost items considered are the capital cost of the power plant, the capital cost of the fossil-fired backup power source for the TUS, the present worth cost of thirty years of fuel and operation and maintenance (O/M), and the capital cost of a cooling tower (sized for peak plant capacity). Table 6.1 lists the cost of each of these items and the total cost for both the absorptive air conditioning

option and the compressive air conditioning option. No fossil-fired backup heat source is listed for the 0% thermal/electric split value because, for this split value, no TUS exists to require backup support. The nuclear power plant is considered to have its own electrical backup system, but electrical backup for the base is supplied by a tie-in to the local utility grid.

Note that power plant capital cost and fuel and O/M cost comprise 95% of the total present worth nuclear option, so that errors in overestimating the cost of the fossil-fired backup or cooling tower oversizing have a small effect on the overall cost total. Detailed calculations of nuclear power plant capital costs are presented in Appendix D.1. In brief, the capital cost of the plant is calculated by evaluating the functional dependence of capital cost versus size, as given by Metcalfe, using the CONCEPT III code [1]. The cost of the fossil-fired backup power plant for the TUS decreases with decreasing values of thermal/electric load split, due to the reduced size of the TUS.

6.3 Coal Gasification Gas Turbine (CGGT) Plant Costs

Section 2.3 lists the important parameters of the CGGT components. Estimating the cost of gasification equipment is complicated by the dependence of equipment cost on coal type and by the reluctance of vendors to commit themselves to statements of unit cost data. Projecting costs into the

TABLE 6.1

NUCLEAR POWER PLANT COST COMPONENTS*

Split	Nuclear Power Plant	Fossil-Fired Backup	Fuel + O/M	Cooling Tower	Total
ABSORPTIVE AIR CONDITIONING:					
100%	102.24	5.95	54.6	2.51	165.3
80%	100.26	4.90	52.9	2.41	160.5
60%	111.90	4.03	58.3	3.00	177.2
40%	138.13	2.54	73.9	4.56	219.1
20%	151.75	1.55	83.0	5.49	241.8
0%	171.11	NA	103.5	6.97	281.6
COMPRESSIVE AIR CONDITIONING:					
100%	122.01	6.15	66.7	3.56	198.4
80%	123.91	5.08	66.0	3.67	198.7
60%	125.35	4.15	69.5	3.76	202.8
40%	136.94	2.61	83.0	4.48	227.0
20%	150.96	1.58	91.1	5.44	249.1
0%	171.11	NA	103.5	6.97	281.6

*In units of 1985 millions of dollars.

future is further complicated by the uncertainty in escalation rates. Escalation rates projected by Metcalfe are used to facilitate comparisons between different power plant types and to insure uniformity between the two reports. The cost of a CGGT system for a given TUS split is determined by matching the capacities of the components to the thermal/electric load calculated by TDIST2. Because the gasifiers and gas turbines are only available in certain sizes, the capital cost of the CGGT plant does not vary continuously with TUS split, but instead changes incrementally as each additional module is added into the system.

Coal consumption is calculated by using the following technique: (detailed in Appendix D.2)

1) A twenty-four hour simulation of the Ft. Knox thermal and electrical power demands for a particular day at a given thermal/electric load split is performed (Figure 5.34),

2) The gas consumption required to generate the electrical demand schedule is calculated by using an average heat rate for the gas turbine generators,

3) The waste heat recovered from the turbine exhaust is subtracted from the total thermal energy demand calculated by TDIST2 — if the total thermal energy demand exceeds the waste heat recovered from the gas turbine additional gas is burned in a central hot water heater,

4) The total amount of coal consumed for the day is found

by adding the electrical gas consumption to any extra heating gas consumption, and converting gas consumption to coal consumption via the gasifier conversion efficiency,

5) The yearly coal consumption for a given thermal/electric load split is found by repeating steps 1 through 4 over the desired range of annual weather variation. This provides the basic data for the annual fuel consumption integration. In practice, an average winter day, an average winter-spring day, an average spring-summer day and an average summer day simulations are used to calculate annual fuel consumption. Fuel consumption is then integrated over the year to obtain total annual fuel consumption. Steps 1 through 5 must be repeated for each thermal/electric split of interest. Additionally, the winter peak and summer peak design day simulations must be performed, since these days determine the TES maximum load and hence the required equipment capacities.

Table 6.2 lists the annual coal consumption for each thermal/electric split value for both the absorptive and compressive air conditioning design options. Note that the relative consumption of fuel (coal) between Option 1 and Option 2 has shifted as compared to the fuel consumption (uranium) for the nuclear power plant. This is directly attributed to the fact that the CGGT plant operates in a load

TABLE 6.2

ANNUAL COAL CONSUMPTION RATES*

Split Value	Option 1 Absorptive Air Conditioning	Option 2 Compressive Air Conditioning
100%	1.90	1.76
80%	1.89	1.81
60%	1.87	1.85
40%	2.03	2.08
20%	2.24	2.29
0%	2.62	2.62

*Coal consumption rates are stated in units of 10^5 tons/year; See Appendix D.3.1 for coal analysis.

following mode whereas the nuclear power plant is constrained to operate in a constant core power mode. Thus, during off-peak demand periods, the coal-fired unit can reduce power and conserve fuel.

Tables 6.3 and 6.4 list, for all plant options, the number of FT4C turbine units and their net costs, the number of Lurgi gasifiers and their net costs, auxiliary boiler costs, the present worth of a thirty year coal supply (at $27/ton), and the present worth of thirty years worth of operating and maintenance costs (at 4 mills/KW(e)hr). It is interesting to note that local minima appear to occur in the total present worth costs at thermal/electric split values of 100% and 60%. This behavior is due to the modular (and therefore incremental cost) nature of the CGGT plant model. That is, the gas turbine and gasifier component costs do not vary uniformly and continuously with thermal electric split values. This is clearly seen in Figure 6.1 where the combined capital cost of the gas turbines and gasifiers is plotted. This effect is reasonable and to be expected. A single FT4C gas turbine has a capacity of 26.3 MW(e); therefore it can handle a range of electrical demand. In moving from one thermal/electric split value to another, the increment in electric demand may not exceed the previous electrical capacity and no additional turbine is added to the system.. Conversely, since the nuclear plant is specially designed for a given case, the unit capacity varies continuously and therefore cost also varies continuously.

TABLE 6.3

COAL GASIFICATION GAS TURBINE PLANT COSTS[1]

ABSORPTIVE AIR CONDITIONING OPTION

Split Value	Turbine Cost	Gasifier Cost	Coal Cost	O/M	Auxiliary Boiler	Total
100%	15.0 (2 units)	30.8 (4 units)	48.4	7.07	6.3	107.6
80%	22.5 (3 units)	38.5 (5 units)	48.1	7.02	3.8	119.9
60%	22.5 (3 units)	38.5 (5 units)	47.6	7.0	1.9	117.5
40%	30.0 (4 units)	46.2 (6 units)	51.8	7.6	(2)	135.5
20%	37.5 (5 units)	53.9 (7 units)	57.0	8.3	(2)	156.7
0%	37.5 (5 units)	69.3 (9 units)	66.6	9.7	(3)	183.2

Notes:

(1) In units of millions of dollars in 1985; see Section 2.1 for unit component costs.

(2) Thermal backup supplied by gasifier boiler capacity.

(3) No TUS requiring backup.

TABLE 6.4

COAL GASIFICATION GAS TURBINE PLANT COSTS(1)

COMPRESSIVE AIR CONDITIONING OPTION

Split Value	Turbine Cost	Gasifier Cost	Coal Cost	O/M	Auxiliary Boiler	Total
100%	22.5 (3 units)	38.5 (5 units)	44.8	6.5	6.8	119.1
80%	30.0 (4 units)	38.5 (5 units)	46.1	6.7	4.3	125.6
60%	30.0 (4 units)	38.5 (5 units)	47.1	6.9	1.9	124.3
40%	30.0 (4 units)	46.2 (6 units)	53.0	7.7	(2)	136.9
20%	37.5 (5 units)	53.9 (7 units)	58.4	8.5	(2)	158.4
0%	37.5 (5 units)	69.3 (9 units)	66.6	9.7	(3)	183.2

Notes:

(1) In units of millions of dollars in 1985; see Section 2.1 for unit component costs.

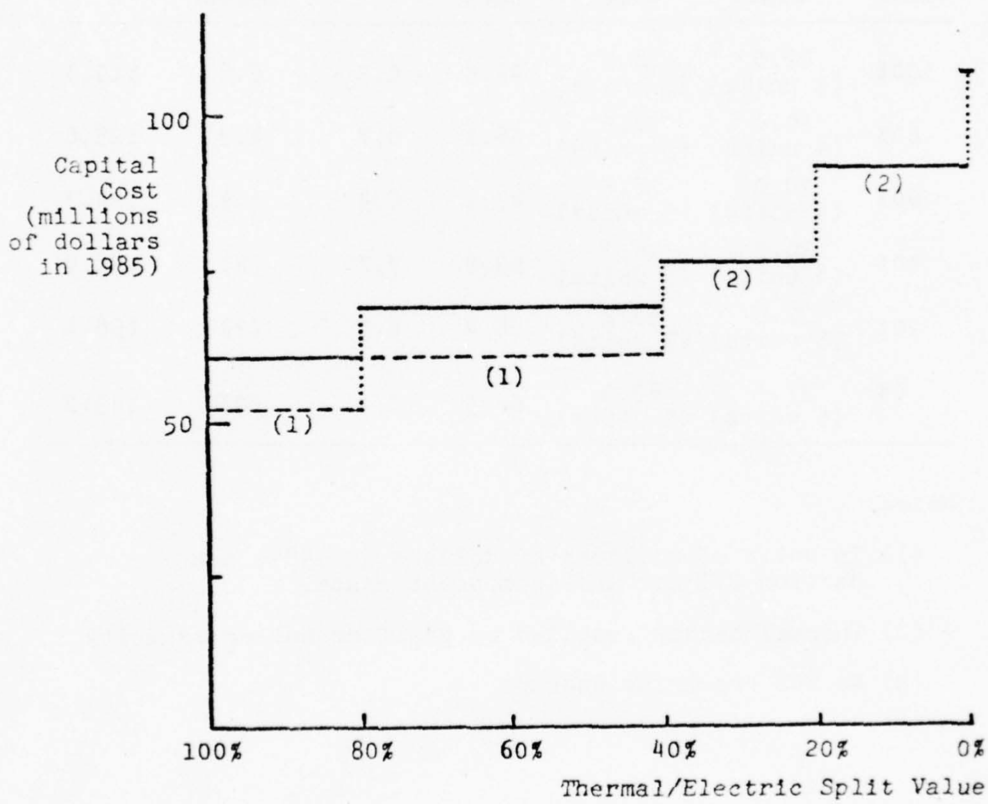
(2) Thermal backup supplied by gasifier boiler capacity.

(3) No TUS requiring backup.

FIGURE 6.1

COMBINED GASIFIER AND TURBINE COSTS

VERSUS THERMAL/ELECTRIC SPLIT VALUE



(1) Absorptive air conditioning option

(2) Same cost for both options

6.4 Thermal Utility System Costs

Thermal Utility System (TUS) costs are composed of the cost of the piping (including insulation and installation), the cost of the pumps, the cost of the heat exchangers and the cost of the thermal storage reservoir. Calculation of the costs for each of these items is detailed in the following sections. Table 6.5 lists the piping costs, the pump costs, the heat exchanger costs, thermal reservoir costs and total costs for each TUS studied.

There are several items to note in Table 6.5. The compressive air conditioning system option pipe costs are lower than the absorptive air conditioning system pipe costs because the absorptive system option secondary piping is larger than the compressive system's secondary piping. The absorptive system secondary piping is larger because it must handle the much higher water flow rates of the chilled water summer cooling loads. Recall that secondary loop heating is supplied by water at 200°F and returning at 80°F, corresponding to a 120°F temperature drop. However, secondary loop cooling is supplied by chilled water supplied at 45°F and returned at 55°F, with only a 10°F temperature change. Therefore, for a given magnitude thermal load, the chilled water system must pump much more water than the heating system. This increased flowrate requirement is reflected in both the pipe and pump costs of the two design options. Heat exchanger costs are virtually identical

TABLE 6.5

THERMAL UTILITY SYSTEM COSTS*

Split	Pipes	Pumps	Heat Exchangers	Reservoir	Total
Absorptive Air Conditioning:					
100%	46.8	.40	6.9	3.7	57.8
80%	37.3	.37	5.8	3.3	46.7
60%	30.3	.31	4.6	3.0	38.2
40%	19.2	.18	2.4	2.4	24.2
20%	12.7	.14	1.7	2.3	16.9
Compressive Air Conditioning:					
100%	33.7	.28	6.9	2.7	43.7
80%	25.9	.22	5.8	2.7	34.5
60%	21.0	.16	4.6	2.6	28.4
40%	13.4	.10	2.4	2.4	18.3
20%	8.6	.07	1.7	2.3	12.7

*In units of millions of dollars in 1985.

since heat exchanger sizes are determined by peak winter loads which are essentially the same for both options.

Thermal storage reservoir costs for the compressive air conditioning option are less than or equal to reservoir costs in the absorptive air conditioning option. Again, this is a direct consequence of the cooling mode selected. Compressive cooling reservoir sizing is based on winter peak load mis-matches rather than summer peak load mis-matches, since the demand on the compressive system TUS during summer is not for cooling but only for the small domestic hot water load. The absorptive cooling reservoir sizing must handle the large thermal mismatches caused by summer-peak air conditioning loads.

A detailed discussion of each cost component in Table 6.5 is given in the following sections.

6.4.1 Piping Costs

The piping specifications and layout of the piping for the TUS is carried out with the guidance of the "Manual of Design Criteria for Military Construction for High Temperature Water Heating Systems with Forced Circulation Boilers," EM 1110-345-162, prepared by Geiringer, P.L. All piping in the system consists of insulated supply and return lines buried in a common trench at a depth of six feet. Table 6.6 lists the net cost of piping, insulation,

TABLE 6.6

THERMAL UTILITY SYSTEM PIPE COST FILE*

Pipe Size (Nominal diameter-inches Schedule 40 pipe)	Cost (Dollars/Foot)
1	28
2	70
3	94
4	98
6	138
8	166
10	209
12	288
14	301
16	397
18	468
20	523
22	570
24	609

*Stated in unit costs in 1985 dollars per linear foot of pipe installed.

excavation and backfill on a unit length basis for the pipe sizes used in the design of the Fort Knox TUS. The base cost data used to produce this table are presented in Appendix D.6 together with listings of length and internal cross-sectional area for each pipe used in each TUS simulation. As suggested by Geiringer, pipe sizing is based on a nominal 8 ft/sec at the pipe's design mass flowrate. The cross-sectional area so calculated will usually lie between two available pipe sizes; the larger pipe is then selected as the design pipe size. Selecting pipe sizes in this manner insures a realistic simulation (i.e., commercially available pipes are used), and it includes a prudent margin for system expansion.

The total cost of piping for a given TUS design is calculated by using the cost table of Table 6.6 together with the pipe listings of Appendix D.6. The TDIST2 simulation checks the pipe cost file to determine the cost per linear foot for a pipe size and multiplies this by the pipe length. In this way, TDIST2 calculates the total cost of the piping used in the simulation.

In practice, however, not every building on a base is explicitly connected to the TUS by a pipe in the TUS simulation. As is discussed in Section 4, buildings are aggregated into load center groups which are used to simulate the building demands. Every building on the base is accounted for with regard to its electrical and thermal

demand, but not all piping is included explicitly in the TUS simulation. In effect, the TUS simulation neglects the pipe that connects an individual building to the water main in the street (which usually is explicitly listed in the TUS simulation). For example, the 100% TUS for Fort Knox lists 122 major pices of pipe. The total number of buildings described in the system is 1499. The cost of the piping required to connect these buildings to the water main in the street is an important, but not a dominating item in the total piping cost. The cost of piping required to connect buildings to the street mains is calculated by correlating pipe length with BTU delivered, as detailed in Appendix D.8. Resulting piping costs are shown in Table 6.5.

6.4.2 Pump Costs

Based on the pipe sizing criterion discussed in Section 6.4.1 and the building loads, the computer simulation calculates the pressure drops experienced by the fluid in the TUS. Pump size is determined by the system design flowrates and pressure drops. Once the pump rating is known, pump costs are evaluated using the correlation presented in Appendix D.7. Appendix D.7 also lists the location and rating of each pump in every TUS option studied. Because the secondary loops are relatively independent of one another, the secondary loop pump ratings do not change significantly from simulation to simulation (of course, secondary

loops disappear as the thermal/electric split value is decreased). However, each thermal/electric split value has a unique primary loop configuration and pump rating resulting from branches of the primary system being deleted as the thermal/electric split value is decreased. Pump costs are shown for each TUS option in Table 6.5.

6.4.3 Heat Exchanger Costs

The TUS heat exchangers used in the TDIST2 simulations are single-pass, counterflow units with 1" O.D. tubes and 1" I.D. channels is square bundle matrices. Appendix C.1 details the design heat transfer coefficients of these heat exchangers and lists their required heat transfer areas and costs. Total heat exchanger costs are also listed in Table 6.5 for each TUS option. The cost function used in computing the capital cost of these heat exchangers — given in Eq. (6.1) — is relatively insensitive to the details of the heat exchanger configuration, requiring only a specification of the total heat transfer area of the unit.

$$C_x = 506{,}000 + 5.9\,A \tag{6.1}$$

where

C_x = heat exchanger cost (dollars in 1985), and

A = total heat transfer area (ft^2).

Although originally derived for 1971 costs [4], the coefficients in Eq. (6.1) have been excalated to 1985 dollars.

Consumer heat exchanger costs (for individual buildings) were estimated by a technique similar to that used to determine building connection pipe costs. As outlined in Appendix C.2, a calculation was performed on a generic consumer heat exchanger to determine the average heat transfer characteristics, in particular the thermal energy delivered per unit heat transfer area. Then, realizing that consumer heat exchangers are low pressure/low temperature heat exchangers, Eq. (6.1) was altered to the form of Eq. (6.2),

$$C_c = 5\,A \tag{6.2}$$

where

C_c = heat exchanger cost (dollars in 1985), and

A = total heat transfer area (ft^2).

It is felt that Eq. (6.2) probably overestimates the costs of the consumer heat exchangers since this equation was originally developed to describe the cost of heat exchangers built to operate under much more severe conditions of temperature and pressure. In any case, inaccuracies in consumer heat exchanger costs would have a small effect on overall TUS cost, since consumer heat exchanger costs make up less than 2% of the total TUS cost.

6.4.4 Thermal Energy Storage Reservoir Costs

In computing the costs of the thermal energy storage reservoir for each system option, the 1973 benchmark cost

of $1 per gallon of storage capacity used by Nida [5] is doubled for conservative estimation and is escalated at 6.2% per year to $4.12 per gallon in 1985. Since the reservoir is composed of an interconnected set of from two to eleven identical tanks, it is assumed that this unit cost applies uniformly to all designs, independently of their total storage volume. Table 6.5 lists the costs of the units considered for installation.

6.5 Electrical Transmission, Distribution and End Use Equipment Costs

Detailed calculations of the costs of electrical transmission, distribution and end use equipment are presented in project reports [4,5]. The calculations of electrical end use equipment costs are performed in a manner similar to the calculation of TUS piping costs. Peak winter and summer weather simulations are run for a given TES option, and the maximum load (heating and cooling) for each building type is used to determine the capacity of the electrical end use equipment for that building type. The total equipment cost is then found by multiplying the unit costs by the number of buildings of that type which are served electrically. For example, for the 100% TUS split value, all buildings would be sized for compressive air conditioning, and consequently the compressive air conditioning cost would

be at a maximum. For intermediate split values (i.e. some buildings heated by TUS hot water, and some heated by heat pumps) those buildings heated by heat pumps are assumed to be cooled by the heat pumps in summer and therefore are not charged the additional cost of compressive air conditioning. Electrical end use equipment is considered to consist of heat pumps, compressive air conditioning, absorptive air conditioning, baseboard electric resistance heaters and electric hot water heaters. The total costs of these components for each TUS option are shown in Table 6.7.

Electrical transmission and distribution costs are made up of the cost of transmission lines, transmission substations, primary distribution lines, distribution substations, line transformers and meters. Transmission and distribution costs listed in Table 6.7 are the marginal costs of that option's transmission and distribution system compared to the transmission and distribution system required to supply the base's electrical non-space conditioning load. Marginal costs rather than total electrical costs are used because the variation of the costs in electrical supply service involves only this factor (i.e. an irreducible minimum electrical service is inescapable, and is reflected in total electrical service costs). The calculation of transmission and distribution costs is detailed in Ref. [4]. It involves correlations of cost depending on total area served, magnitude of the demand and the type of consumer.

TABLE 6.7

ELECTRICAL END USE EQUIPMENT, MARGINAL

TRANSMISSION AND DISTRIBUTION COSTS*

Split	Heat Pumps	Air Conditioning	Electric Hot Water	Transmission & Distribution	Total
Compressive Air Conditioning Option:					
100%	NA	3.45	0	8.18	11.63
80%	10.59	2.86	.049	8.52	22.02
60%	19.41	2.33	.0996	8.77	30.61
40%	35.15	1.44	.1495	11.53	48.27
20%	44.42	.721	.2055	13.68	59.03
0%	60.03	0	.228	16.86	77.12
Absorptive Air Conditioning Option:					
100%	0	10.90	.228	5.18	16.31
80%	10.59	9.35	.228	7.18	27.35
60%	19.41	8.18	.228	8.90	36.72
40%	35.16	5.15	.228	11.94	52.48
20%	44.41	3.71	.228	13.94	62.29

*Costs stated in units of millions of 1985 dollars.

The cost of the base case electrical transmission and distribution system (supplying only non-space conditioning demand) was calculated to be 55.9 million dollars in 1985.

6.6 TES Cost Minimization

Tables 6.8-6.9 list the costs of the central nuclear power station (HTGR), the coal gasification-gas turbine plant (CGGT), the Thermal Utility System (TUS), electrical equipment (E) and total cost for each TES option. The total TES cost is found by adding the component costs of the system. Fig. 6.2 shows the TES costs as a function of thermal/electric split value. It should be noted that the discontinuities in the CGGT-TES cost curves are due to incremental additions of either turbine or gasifier units.

Not surprisingly (for these plant sizes), the CGGT-TES is everywhere less expensive than the HTGR-TES for a unit coal cost of $27/ton. The breakeven coal cost for the compressive air conditioning CGGT-TES compared to the compressive air conditioning HTGR-TES is $69.8/ton, and remains near this value over the range of split values. The breakeven coal cost for the absorptive air conditioning CGGT-TES compared to the absorptive air conditioning HTGR-TES is $45.6/ton at a split value of 80%.

The compressive and absorptive air conditioning option cost curves converge at low thermal/electric split values because as the split value decreases both options closely

TABLE 6.8

TOTAL ENERGY SYSTEM COSTS*

ABSORPTIVE AIR CONDITIONING OPTION

Split	HTGR	CGGT	TUS	E	Total
100%	165.3		57.79	16.31	239.4
		107.6	57.79	16.31	181.7
80%	160.5		46.66	27.35	234.5
		119.9	46.66	27.35	193.9
60%	177.2		38.22	36.72	252.1
		117.5	38.22	36.72	192.4
40%	219.1		24.15	52.48	295.7
		135.5	24.15	52.48	212.1
20%	241.8		16.88	62.29	321.0
		156.7	16.88	62.29	235.9
0%	281.6		0	77.12	358.7
		183.2	0	77.12	260.3

*Costs stated in units of millions of 1985 dollars.

TABLE 6.9

TOTAL ENERGY SYSTEM COSTS*

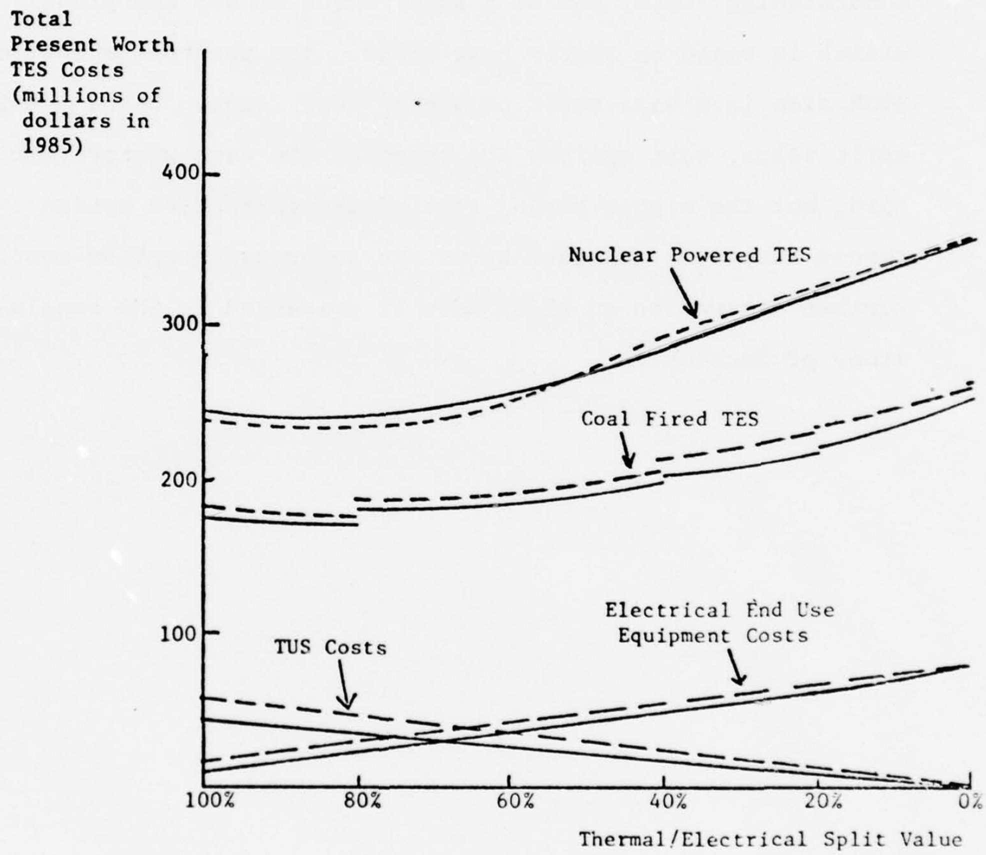
COMPRESSIVE AIR CONDITIONING SYSTEM

Split	HTGR	CGGT	TUS	E	Total
100%	198.4		43.67	11.63	253.7
		119.1	43.67	11.63	174.4
80%	198.7		34.53	22.02	255.3
		125.6	34.53	22.02	182.2
60%	202.8		28.36	30.61	261.8
		124.3	28.36	30.61	183.3
40%	227.0		18.25	48.27	293.5
		136.9	18.25	48.27	203.4
20%	249.1		12.71	59.03	320.8
		158.4	12.71	59.03	230.1
0%	281.6		0	77.12	358.7
		183.2	0	77.12	260.3

*Costs stated in units of millions of 1985 dollars.

FIGURE 6.2

TOTAL ENERGY SYSTEM COSTS*



*Solid lines indicate Compressive Air Conditioning Option; dashed lines indicate Absorptive Air Conditioning Option

resemble each other. The crossing of the HTGR-powered TES curves between 60% and 40% thermal-electric split values is due to the shift in the sizing requirement of the compressive air conditioning option plant. At high split values, the compressive option plant size is based on summer peak air conditioning loads, but at a split value of 40% the plant sizing is based on winter peak loads. The absorptive option HTGR size is always based on winter peak loads. Below a 40% split value, both options are based on the same winter peak load, but the higher piping cost of the absorptive option forces its total TES cost above the compressive option cost. Further discussion on these data is presented in the conclusions of Section 7.

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CHAPTER 7

CONCLUSIONS AND RECOMMENDATIONS

7.1 Conclusions

From the economic analysis of Chapter 6, it is seen that the global minimum cost TES occurs at a thermal/electric split value of 100% for the coal powered compressive air conditioning option. Further consideration of the coal consumption and capital cost curves of Chapter 6 leads to the conclusion that the actual minimum lies somewhere between the 80% and 100% split values, at the point where base electrical demand is just satisfied by the three FT4C turbines of the 100% split and coal consumption has been reduced by optimizing the consumer end use-equipment mix. Practically, this is an advantageous characteristic because it indicates that cost reductions can be achieved by shedding distant loads from the TUS (thereby reducing TUS costs) and shifting the loads to electrical power (optimizing the equipment mix) thereby reducing annual coal consumption. This of course is the reason for which various thermal/electric split values are analyzed.

It is interesting to note in Tables 6.6-6.7 and Fig. 6.2 that the compressive air conditioning TES option exhibits the lowest cost over life. Intuitively it would seem that absorptive air conditioning, utilizing the waste heat available due to electrical production would give the lowest costs. However, this is not the case. For almost every item, the

comparative cost of the compressive versus the absorptive system favors the compressive system; however, the cost differences between the two systems are small. From tables in Chapter 6, the major penalties associated with the absorptive air conditioning system compared to the compressive system are the increased cost of secondary pipe required in the absorptive air conditioning TUS secondary loops (compared to compressive air conditioning TUS secondary loops) and the increased cost of absorptive air conditioning units compared to compressive air conditioners (on a unit capacity basis). The only significant cost items which penalize the compressive air conditioning option are the increased costs of plant capacity and the electrical transmission and distribution system. It should be emphasized that the cost margin separating the absorptive air conditioning system cost from the compressive air conditioning system cost is too small to justify unequivocally the preference of one option over the other. Secondary characteristics of either option may be the deciding factors in system selection, or future changes in component capital costs could change the result.

Compared to a similar TES analysis studying Fort Bragg, N.C. [1], the optimal thermal/electric split value has been increased slightly from a 75% thermal/electric split value to an 80% split value. This shift is due to including electrical transmission, distribution and end use equipment costs in the TES costs. The Fort Bragg analysis assumed this cost

to be approximately independent of split value. Calculations have now shown that the actual electrical system costs increase as split value is reduced. This tends to shift the minimum of the total cost curve to higher split values, explaining the increase in the optimum split value. It should be emphasized that, in general, the optimum split value will not be in the 75% to 80% range. The optimum split value depends on the climate at the site and the operating characteristics of the end use equipment. For example, extreme northern sites would be unable to use heat pumps profitably and therefore the optimum split value would probably be near 100%. Alternatively, the split value would change if portions of a base were selectively excluded from some service. For example, if air conditioning were restricted to administration, training and senior officer homes, this would change secondary loop pipe sizes and shift the optimum split value. Similar optimum values of thermal/electric split for Fort Bragg and Fort Knox were expected since end use equipment specification and climate are similar for both sites.

The economic data of Chapter 6 and the Appendices may be used for approximate analyses of alternative combinations of end use equipment. For example, the approximate cost of a TES supplying only heat, non-space conditioning electricity and domestic hot water may be found by summation of the costs of the absorptive air conditioning sized central plant, the costs of the compressive air conditioning sized TUS, the costs

of electrical end-use equipment sized for absorptive air conditioning (subtracting the capital costs of the absorptive air conditioners) and adding electrical transmission and distribution costs for the absorptive system. Using this technique, rough survey calculations may be performed of other interesting possible TUS configurations as the interest arises.

Using total energy systems to supply a base's thermal and electrical needs offers an opportunity to conserve valuable fuel resources. However, Table 7.1 shows that, with the coal gasification-gas turbine equipment commercially available today, the most efficient energy conservation strategy will not be realized. Table 7.1 lists the annual total of electrical and thermal demands supplied by the CGGT-TES, together with the annual coal consumption of the CGGT-TES. Also shown is the coal consumption which would occur if electricity for the base were supplied by a 33%-efficient coal fired electric power plant, and with coal also being consumed on the base to supply the required base TUS demand. The central coal-fired electric plant is assumed to operate without waste heat recovery, and the base's coal-fired boiler is assumed to have an efficiency of 80%. The equivalent combined efficiency for electrical production of the coal gasification-gas turbine plant is only 20%. Thus, although the CGGT plant recovers energy from the turbine exhaust for the TUS, the overall CGGT efficiency is so low that increased coal consumption results

TABLE 7.1

COMPARISON OF ANNUAL COAL CONSUMPTION RATES FOR THE CGGT-TES

AND FOR

AN ELECTRIC UTILITY AND ON-BASE THERMAL BOILER SYSTEM

	Annual Electrical Energy Demand (Units of 10^5MWe-hr)	Annual Thermal Energy Demand (Units of 10^5MWt-hr)	Annual Coal Consumption (Units of 1000 tons)
Compressive Air Conditioning Option 80% TUS	1.45	2.12	181
Conventional Utilities*	1.45		78.9
		2.12	47.6
Total Coal Consumption			127.

*Assumes coal heating value of 9,500 Btu/lb; electrical generating efficiency of 33% and boiler efficiency of 80%.

from its use. This point requires further discussion. It must be emphasized that usually total energy systems are expected to save energy compared to separate electrical and thermal energy production, as in the case of diesel engines driving electric generators with heat recovery from the diesel exhaust. This system burns diesel oil and does offer high energy efficiency. The coal gasification-gas turbine plant discussed in this report begins with a solid fuel and converts it to a gas. The gas is then used in a turbine generator set including waste heat recovery from the turbine exhaust. Note, however, that the overall cycle must now include the gasification energy conversion efficiency. For this study an efficiency of 70% was used. [2] A turbine average electrical generation efficiency of 25% was assumed to apply. The maximum electrical conversion efficiency for the FT4C turbine is 27% [3]. Thus, parameters used in this study fall within the nominal range of expected component performance. The final answer must be that CGGT technology is not yet sufficiently advanced to offer fuel savings in the type of mixed energy product required for a total energy system.

It may be that a coal fired Rankine cycle plant using a back pressure turbine or a steam extraction turbine could supply base energy demands at higher net efficiencies, but this would require further study. Such a system would be able to combust coal directly, but would have the additional

problems of sulfur and particulate removal from stack gases.

Herein lies the desirability of using a coal gasification-gas turbine central station. The coal gasification process allows easier and more convenient and more efficient removal of sulfur and particulates from the product gas. Removing sulfur and particulates from the product gas is necessary to protect the internals of the system's gas turbines. However, it automatically insures that turbine exhaust will meet sulfur and particulate clean air standards. In addition, the gasification process produces elemental sulfur, rather than the obnoxious sulfur sludge produced by conventional stack gas scrubbers.

Table 7.2 lists the cost of electricity for the 80% TUS compressive air conditioning option. This cost of electricity is based on a nuclear central plant cost of $132.68 million (1985 dollars), annual fuel plus O/M costs of $7.00 million (1985 dollars) and a 10% cost of money, giving a busbar energy cost of 140 mills/KW(e)hr. Also shown in Table 7.2 is the equivalent cost of gas, based on a gasifier cost of $38.5 million (1985 dollars), annual fuel plus O/M costs of $5.59 million (1985 dollars) and a 10% cost of money, giving a levelized gas cost of 392 cents per million Btu.

Figure 7.1 shows the capital cost of the nuclear-powered and coal-fired total energy systems, replotted from Fig. 6.2. The error bands associated with the coal-fired option indicate the effect of changing coal prices. The base

TABLE 7.2

EQUIVALENT COSTS OF GAS AND ELECTRICITY(1)

FOR THE CGGT TES

HTGR Capital Cost(2)	\$132.7 million
Annual Fuel and O/M Costs	\$7.00 million
Electrical Production	145 million KW(e)hr
Levelized Cost of Electricity	140 mills/KW(e)hr
Gasifier Capital Costs(3)	\$38.5 million
Annual Fuel and O/M Cost	\$5.59 million
Annual Gas Production Rate	2.41×10^{12} Btu
Levelized Cost of Gas	392 units per million Btu

Notes:

(1) Costs are based on the 80% TUS compressive air conditioning option; all costs stated in 1985 dollars; and 10% cost of money.

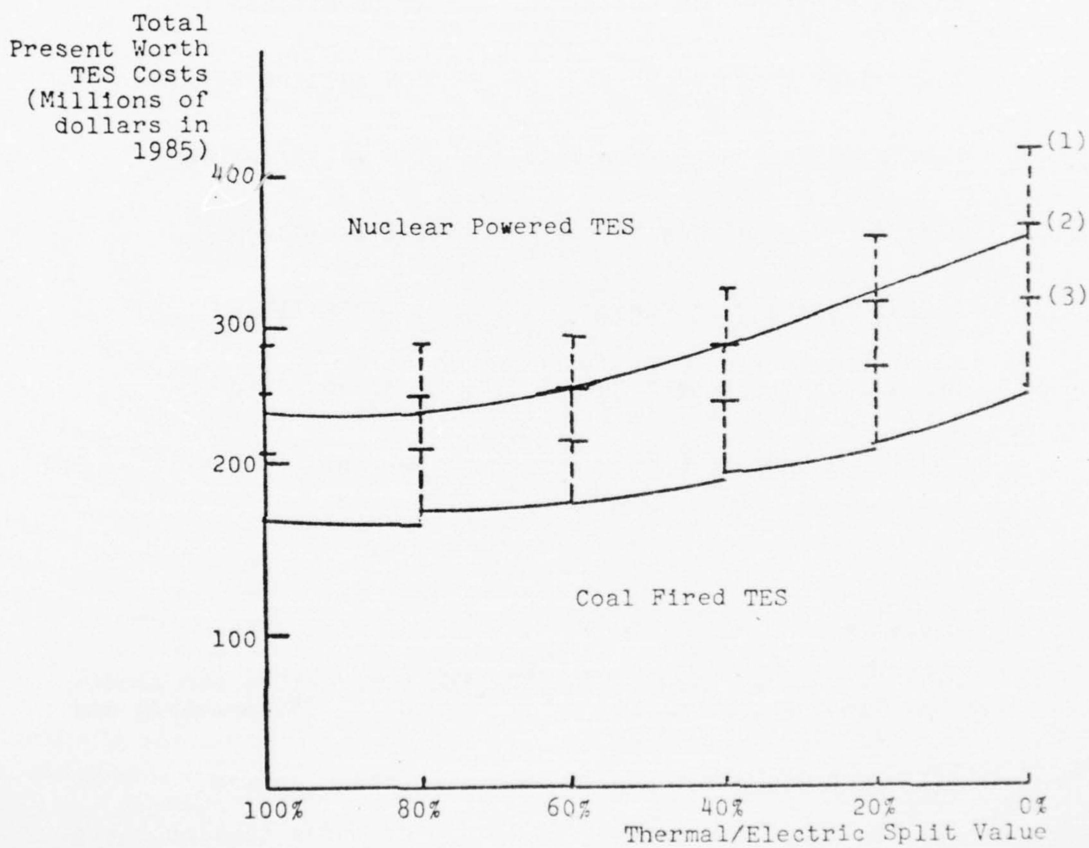
(2) HTGR sized for peak summer electrical demand.

(3) Includes cost of gasifiers but excludes turbine costs.

FIGURE 7.1

EFFECT OF INCREASED COAL COSTS ON THE COMPARATIVE

COSTS OF NUCLEAR AND COAL FIRED TOTAL ENERGY SYSTEMS



Notes:
(1) Based on coal cost of 90 (dollars/ton)
(2) Based on coal cost of 70 (dollars/ton)
(3) Based on coal cost of 50 (dollars/ton)

case is calculated for a coal price of $27 per ton, but the effects of $50 per ton and of $70 per ton coal are shown. Thus the breakeven cost of coal may be seen conveniently at any thermal/electric split value.

Figure 7.2 shows the sensitivity of total system cost to variations in the base case component cost. The parameter plotted is given by Equation (7.1)

$$z = \frac{\dfrac{\Delta \text{ Total Cost}}{\text{Total Cost}}}{\dfrac{\Delta \text{ Component Cost}}{\text{Component Cost}}}, \qquad (7.1)$$

where

Total Cost = total present-worth cost of a given TES option

Component Cost = present-worth cost of a given component (e.g. the TUS cost)

Δ Component Cost = the assumed change in component cost

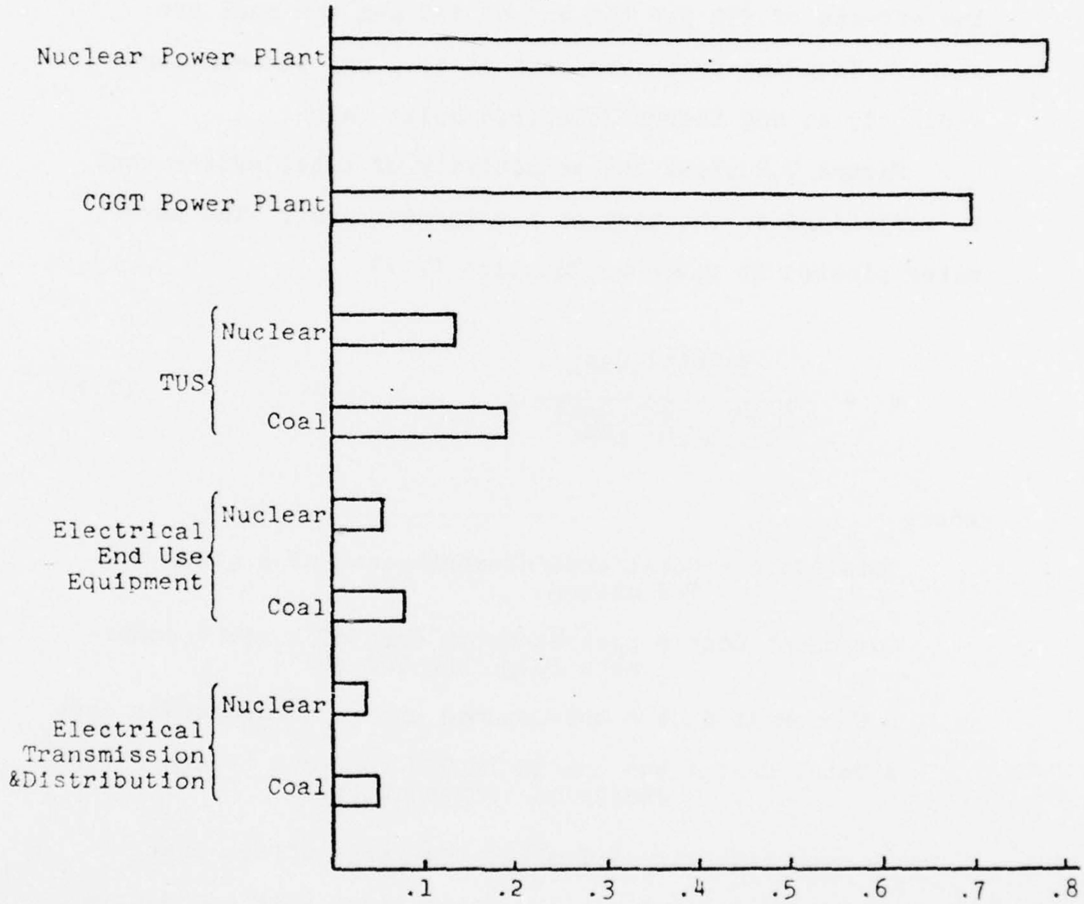
Δ Total Cost = the change in TES cost due to the given change in component cost.

As expected, Fig. 7.2 shows the large effect that changes in the central station power plant cost would have on overall TES costs. A surprising feature of this histogram is the relatively small effect played by electrical end use equipment.

FIGURE 7.2

SENSITIVITY OF TOTAL COSTS(1) TO VARIATIONS(2)

IN COMPONENT COSTS



Notes:

(1) The figure shows the sensitivity of total costs to changes in component cost for the 80% TUS compressive air conditioning option; both the nuclear and coal fired options are analyzed.

(2) Assumes a 20% change in base cost of component.

7.2 Recommendations

The final selection of a power source for the Fort Knox TES must be based on important criteria in addition to expected monetary cost. Within the accuracy of the calculations, the optimal CGGT TES cost is 182 million dollars (1985), which is much less expensive than the optimal nuclear option. This advantage could be easily negated by rapidly rising coal costs. Alternatively, cost overruns on the nuclear power plant could significantly increase the cost of the nuclear option. In addition, each system has important secondary characteristics which argue in favor of its selection for TES use.

Some of the more important power plant selection tradeoffs are summarized in Table 1.1. For example, the CGGT system has the advantage of modular add-on potential growth. It is relatively easy to install another gasifier or gas turbine unit to the power plant as is required by the expansion of the base TES. The nuclear power plant is limited severely in its add-on growth ability.

The nuclear plant does have a significant advantage in the dependability of its fuel supply. A freshly fueled HTGR would be expected to supply from 3 to 6 years of service before refueling would be required. Conversely, storage of more than a few months' supply of coal on base seems impractical because of the large bulk of such a coal pile.

The ultimate selection of the power plant type must be based on the user's present and projected needs not only for electrical and thermal power, but also for personnel with experience in new technologies. As directed by the President, the Department of Defense may wish to gain operating experience with coal gasification technologies. As coal comes into increased use, the favorable sulfur removal characteristics of the coal gasification process may become a mandated part of the solution to the stack gas clean-up problem.

Survey calculations of TUS pipe costs were performed using plastic pipe in the low temperature and low pressure secondary loops. Plastic pipe data was obtained from an Oak Ridge National Laboratory report [4]. Steel pipe (listed in the 100% TUS absorptive air conditioning option) in the secondary loops costs 22.6 million dollars (1985). If plastic pipe could be used, the corresponding cost would be 7.2 million dollars (1985). Overall pipe cost savings are not as dramatic. The all-steel system costs 36 million dollars (1985), while the steel and plastic system costs 20.6 million dollars (1985). Although plastic pipe is not yet authorized by the DOD for heating hot water distribution systems, the potential savings are so large that continued interest is warranted.

In conclusion, it is felt that the TES analyses presented in this report include all significant physical phenomena and

and economic factors pertinent to the analysis of mixed product total energy systems. It is felt that the economic analyses presented in this report are based on reasonable assumptions and should provide useful estimates of anticipated costs.

REFERENCES

1. Stetkar, J.W., Best, F.R., Golay, M.W., "Design of a Nuclear-Powered Total Energy System for Ft. Bragg, North Carolina," Final report under Contract No. DAAK02-74-C-0308, MIT, Department of Nuclear Engineering, May 1976.

2. Boyd, W.D., Golay, M.W., "Economic and Technical Aspects of Coal Gasification for Use in Gas Turbine Operation," Project Report, Contract No. DAAK02-74-C-0308, MIT, Department of Nuclear Engineering, 1976.

3. Kelly, J., Golay, M.W., "Economic and Technical Aspects of Gas Turbine Power Stations in Total Energy Applications," Project Report, Contract No. DAAK02-74-C-0308, MIT, Department of Nuclear Engineering, 1976.

4. Meador, J.T., "MIUS Technology Evaluation - Thermal Energy Conveyance," Report No. ORNL/HUD/MIUS-22, UC-38, Contract No. W-7405-eng-26, Oak Ridge National Laboratory, May 1976.

APPENDIX A.1

FUEL SAVINGS ACHIEVED BY A CENTRAL STATION TUS COMPARED TO CONVENTIONAL HEATING

Central Station Gas-Fired Heater Efficiency

$$= \frac{\text{BTU absorbed by water}}{\text{BTU of fuel consumed}} = 70\% \quad [6]$$

Individual "Home" Gas-Fired Heater Efficiency

$$= \frac{\text{BTU absorbed by water}}{\text{BTU of fuel consumed}} = 40\% \quad [6]$$

Average fraction of thermal load recovered by Turbine Exhaust Waste heat exchangers = 82.5% for 80% thermal split absorptive air conditioning option.

Assume heat load of 100 units

1. "Home" Heaters would require

$$\frac{100}{40\%} = 250 \text{ units}$$

2. Central Station TUS would require

$$\frac{100(1 - .825)}{70\%} = 25 \text{ units}$$

Thus the Central Station TUS reduces fuel consumption by

$$\frac{225 \text{ units}}{250 \text{ units}} = 90\%$$

APPENDIX A.2

COST OF ENERGY STORAGE AS HOT WATER COMPARED TO GAS STORAGE

I. Gas Storage at 300 psi costs $\$12/ft^3$

For our system gas H.V. = 125 BTU/SCF

Cost of Energy Storage as Gas at 300 psi =

$$\frac{\$12}{ft^3} \; \frac{SCF}{125\ BTU} \; \frac{ft^3}{0.49\ SCF} = \frac{\$1.96}{BTU}$$

II. Hot Water Storage $\$7.5/ft^3$

Energy stored in each ft^3 =

$$\frac{1\ BTU}{lbm^\circ F}\,(380^\circ F - 150^\circ F)\,\frac{lbm}{.017 ft^3} = 13529\ BTU/ft^3$$

Cost of energy storage as hot water =

$$\frac{\$7.5}{ft^3}\left[\frac{1}{13529\ \frac{BTU}{ft^3}}\right] = \frac{\$.00055}{BTU}$$

Clearly it is less expensive to store energy as hot water than gas.

APPENDIX B

FORT KNOX CONSUMER SPECIFICATIONS

In order to be able to compute the conduction, solar incidence, ventilation and internal heat generation components of the space conditioning demands and the domestic hot water usage for the specified energy consumer categories, TDIST requires the following data for each building type to be analyzed: the explosed areas and thermal resistances of walls, windows, the roof and the basement; the building height; its orientation; the outer wall and roof surface materials; the wall and roof solar absorptivities; a composite internal room and glass material window shading coefficient; the shading of each wall and the roof; the nominal maximum desired ventilation air flow rate; the total connected electrical load in the building, exclusive of any electrical space conditioning equipment; the maximum rate of domestic hot water usage; crack lengths and flow coefficients for openings around doors and windows and cracks in the structural walls; a desired internal room temperature profile to be maintained by the space conditioning equipment throughout the analysis period; a schedule of building use factors relating appliance, lighting and ventilation requirements to the building occupancy; and a schedule of domestic hot water usage.

Because much of these data depend strongly upon t precise nature of the building types chosen to be analy Army personnel were requested to supply as much of the formation as possible for each of the representative Fo Knox building categories. Members of the project from together with FESA representatives, conducted an inform gathering visit to Fort Knox. This visit greatly facil tated identification of building types on the base and allowed estimates to be made of such intangible factors as building and window shading coefficients.

Army personnel lifestyles, military energy usage, conservation regulations expected to be in force in 198 were used in establishing building use schedules and space conditioning indoor temperature requirements. In particular, special care was taken in specifying the building shading coefficients, infiltration air flow co ficients, ventilation requirements and building use pro files, since it has been found that solar radiational h ing and the combined effects of infiltration and forced air ventilation air flows contribute significantly to t total building space conditioning loads. Large variati in these coefficients in the available literature, a ge lack of detailed information from direct field measurem and the individual building-specific nature of such fac as tree shading and window weatherstripping made the ta of formulating these specifications for a "typical" bui

unit especially difficult.

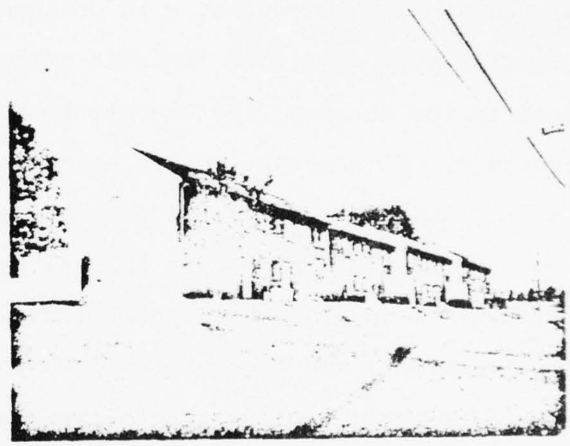
Verification of TDIST2 load calculations was perfo by comparison of TDIST2 output with measured data for a large multi-family dwelling. Details of this analysis w presented in the October 1976 Monthly Progress Report un this contract. In general, TDIST2 calculations were sho to be within 10%-15% of measured data and of the same accuracy as NBSLD [3] and E-Cube [2] calculations.

Tables B.1-16 and associated figures present the f data specifications and assumptions made for each of the fifteen consumer categories. Photographs of some of the most generalized building categories, such as the wareho or community center, are not available since no single building is sufficiently representative.

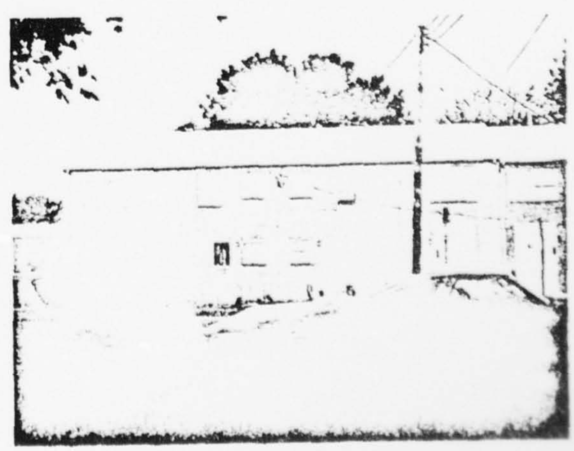
Figure B.1

Type 1: Family Housing

Unit: Duplex, Large



Building number 7720



Building number 7486

AD-A043 701 MASSACHUSETTS INST OF TECH CAMBRIDGE DEPT OF NUCLEAR--ETC F/G 10/2
ANALYSIS OF NUCLEAR AND COAL FUELED TOTAL ENERGY SYSTEM OPTIONS--ETC(U)
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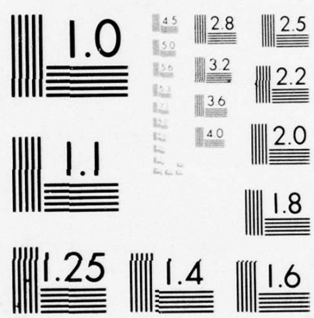

1.0
1.1
1.25
1.4
1.6
1.8
2.0
2.2
2.5
2.8
3.2
3.6
4.0
4.5
5.0
5.6

TABLE B.1

BUILDING INPUT DATA

Type 1: Family Housing

Unit: Duplex, Large

Wall Area: 3080 ft^2

Wall Composite Thermal Resistance: 3.4 $hr \cdot ft^2 \cdot {}^oF/BTU$

Window Area: 1340 ft^2

Window Thermal Resistance: .89 $hr \cdot ft^2 \cdot {}^oF/BTU$
Single pane, no storm windows

Roof Area: 3000 ft^2

Roof Composite Thermal Resistance: 16.67 $hr \cdot ft^2 \cdot {}^oF/BTU$

Basement Ground Contact Area: 3000 ft^2

Basement Wall Thermal Resistance: 4.91 $hr \cdot ft^2 \cdot {}^oF/BTU$

Building Height: 17 ft

Building Orientation: 45^o from North

Wall Surface Material: Brick

Roof Surface Material: Asphalt Shingle

Wall Solar Absorptivity: .70

Roof Solar Absorptivity: .80

Window Shading Coefficient: .50 (50% of incident radiation transmitted

Wall Fraction Lit: .90

Roof Fraction Lit: 1.00 (no shading)

Door Crack Length: 120 ft

Door Air Flow Coefficients: C:40 N:0.50 (see Table B.16)

Window Crack Length: 280 ft

TABLE B.1 (Continued)

Window Air Flow Coefficients: C: 1.7 N: 0.66 (see Table B.16)

Wall Air Flow Coefficients: C: 0.01 N: 0.80 (see Table B.16)

Peak Ventilation: 600 CFM — assumed one air change per hour as per Army measurements and typical residential data

Connected Electrical Load: 3.66 KW, primarily lighting at 0.87 watts/ft^2 total floor area

Peak Domestic Hot Water Demand: 26,000 BTU/hr
Assumed peak of 48 gal/hr for a total of 16 adults

Winter Room Temperature: 68°F (minimum)

Summer Room Temperature: 75°F (maximum)

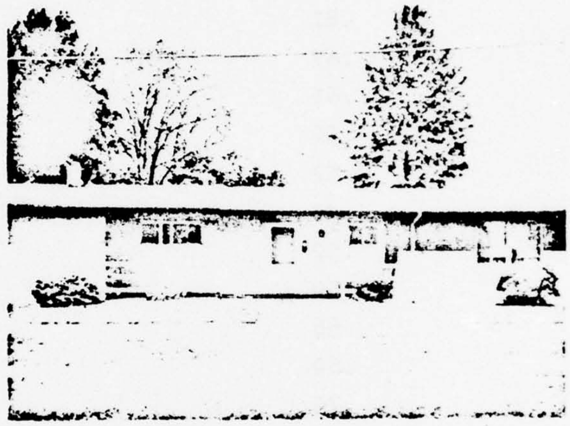
TABLE B.1 (Continued)

Time	Building Use Factor (for electrical equipment and ventilation	Domestic Hot Water Use Factor (ref. ASHRAE)
12	.81	.17
1 AM	.67	.14
2	.61	.13
3	.58	.10
4	.52	.11
5	.49	.10
6	.52	.10
7	.59	.13
8	.66	.15
9	.69	.25
10	.79	.21
11	.90	.19
12	.93	.17
1	.96	.18
2	.96	.15
3	.93	.13
4	.95	.12
5 PM	.93	.12
6	.98	.15
7	1.00	.19
8	.99	.21
9	.96	.18
10	.93	.15
11	.87	.13
12	.81	.17

Figure B.2

Type 2: Family Housing

Unit: Two Family Brick Duplex



Building number 4364

Building number 4364

TABLE B.2

BUILDING INPUT DATA

Type 2: Family Housing

Unit: Two Family Brick Duplex

Wall Area: 4784 ft^2

Wall Composite Thermal Resistance: 3.7 $hr \cdot ft^2 \cdot {}^oF/BTU$

Window Area: 330 ft^2

Window Thermal Resistance: 0.89 $hr \cdot ft^2 \cdot {}^oF/BTU$
Assumed single pane, no storm windows

Roof Area: 1519 ft^2

Roof Composite Thermal Resistance: 16.67 $hr \cdot ft^2 \cdot {}^oF/BTU$

Basement Ground-Contact Area: 1824 ft^2

Basement Wall Thermal Resistance: 4.91 $hr \cdot ft^2 \cdot {}^oF/BTU$

Building Height: 13 ft

Building Orientation: 45^o from North

Wall Surface Material: Brick

Roof Surface Material: Asphalt Shingle

Wall Solar Absorptivity: 0.70

Roof Solar Absorptivity: 0.80

Window Shading Coefficient: 0.45 (45% of incident radiation transmitted)
Assumed blinds or drapes as in typical residences

Wall Fraction Lit: 0.70 (30% of each wall shaded)
Assumed, based on photographs of Fort Knox housing

Roof Fraction Lit: 1.00 (no shading)

TABLE B.2 (Continued)

Door Crack Length: 31 ft

Door Air Flow Coefficients: C: 40 N: 0.50
See Table B.16

Window Crack Length: 195 ft

Window Air Flow Coefficients: C: 1.7 N: 0.66
See Table B.16

Peak Ventilation: 702 CFM
Assumed one air change per hour as per Army measurements and typical residential data

Connected Electrical Load: 3.17 KW
Primarily lighting at 0.87 watts/ft^2 total floor area

Peak Domestic Hot Water Demand: 13,018 BTU/hr
Assumed peak of 24 gal/hr to serve two families as per ASHRAE Systems, 1973 [1]

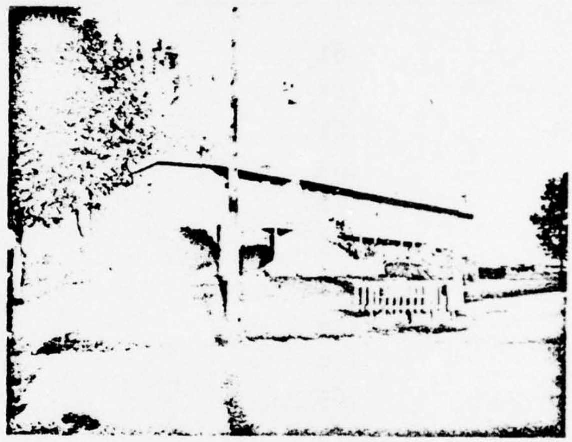
Winter Room Temperatures: 68^oF (minimum)

Summer Room Temperatures: 75^oF (maximum)

TABLE B.2 (Continued)

Time	Building Use Factor (for electrical equipment and ventilation	Domestic Hot Water Use Factor (from Ref. 1 for Apartments)
12	.81	.17
1 AM	.67	.14
2	.61	.13
3	.58	.10
4	.52	.11
5	.49	.10
6	.52	.10
7	.59	.13
8	.66	.15
9	.69	.25
10	.79	.21
11	.90	.19
12	.93	.17
1 PM	.96	.18
2	.96	.15
3	.93	.13
4	.95	.12
5	.93	.12
6	.98	.15
7	1.00	.19
8	.99	.21
9	.96	.18
10	.93	.15
11	.87	.13
12	.81	.17

Building B.3
Type 3: Family Housing, Row
Unit: Four Family Dwelling



Building number 7750

Building number 857

TABLE B.3

BUILDING INPUT DATA

Type 3: Family Housing, Row

Unit: Four Family Dwelling

Wall Area: 5250 ft^2

Wall Composite Thermal Resistance: 4.00 $hr \cdot ft^2 \cdot {}^\circ F/BTU$

Window Area: 532 ft^2

Window Thermal Resistance: 0.89 $hr \cdot ft^2 \cdot {}^\circ F/BTU$
Assumed single pane, no storm windows

Roof Area: 3664 ft^2

Roof Composite Thermal Resistance: 4.55 $hr \cdot ft^2 \cdot {}^\circ F/BTU$

Basement Ground-Contact Area: 3750 ft^2

Basement Wall Thermal Resistance: 4.91 $hr \cdot ft^2 \cdot {}^\circ F/BTU$

Building Height: 17.5 ft

Building Orientation: 45° from North

Wall Surface Material: Brick

Roof Surface Material: Asphalt Shingle

Wall Solar Absorptivity: 0.70

Roof Solar Absorptivity: 0.80

Window Shading Coefficient: 0.50 (50% of incident radiation transmitted)
Assumed use of blinds or drapes as in typical residences

Wall Fraction Lit: 0.90 (10% of each wall shaded)
Assumed, based on photographs of Fort Knox housing

Roof Fraction Lit: 1.00 (no shading)

TABLE B.3 (Continued)

Door Crack Length: 132 ft

Door Air Flow Coefficients: C: 40 N: 0.50
See Table B.16

Window Crack Length: 306 ft

Window Air Flow Coefficients: C: 1.7 N: 0.66
See Table B.16

Wall Air Flow Coefficients: C: 0.004 N: 0.70
See Table B.16

Peak Ventilation: 1103 CFM ·
Assumed one air change per hour as per Army measurements and typical residential data

Connected Electrical Load: 6.53 KW
Primarily lighting at 0.87 watts/ft^2 total floor area

Peak Domestic Hot Water Demand: 39,055 BTU/hr
Assumed peak of 72 gal/hr to serve up to six families as per ASHRAE Systems 1973 [1] .

Winter Room Temperatures: 68°F (minimum)

Summer Room Temperatures: 75°F (maximum)

TABLE B.3 (continued)

Time		Building Use Factor (for electrical equipment and ventilation)	Domestic Hot Water Use Factor (from Ref.1 for Apartments)
12		.81	.17
1		.67	.14
2		.61	.13
3		.58	.10
4	AM	.52	.11
5		.49	.10
6		.52	.10
7		.59	.13
8		.66	.15
9		.69	.25
10		.79	.21
11		.90	.19
12		.93	.17
1		.96	.18
2		.96	.15
3		.93	.13
4	PM	.95	.12
5		.93	.12
6		.98	.15
7		1.00	.19
8		.99	.21
9		.96	.18
10		.93	.15
11		.87	.13
12		.81	.17

Figure B.4
Type 4: Family Housing
Unit: Single Family Detached Dwelling

Building number 1403

Building number 1403

TABLE B.4

BUILDING INPUT DATA

Type 4: Family Housing

Unit: Single Family Detached Dwelling

Wall Area: 5244 ft^2

Wall Compsite Thermal Resistance: 3.45(hr)(ft^2)(°F)/BTU

Window Area: 904 ft^2

Window Thermal Resistance: 0.89(hr)(ft^2)(°F)/BTU
•Assumed Single pane, no storm windows

Roof Area: 2325 ft^2

Roof Composite Thermal Resistance: 16.67(hr)(ft^2)(°F)/BTU

Basement Ground-Contact Area: 2074 ft^2

Basement Wall Thermal Resistance: 4.91(hr)(ft^2)(°F)/BTU

Building Height: 28.5 ft

Building Orientation: 45° from North

Wall Surface Material: Brick

Roof Surface Material: Asphalt Shingle

Wall Solar Absorptivity: 0.70

Roof Solar Absorptivity: 0.80

Window Shading Coefficient: 0.50(50% of incident radiation transmitted)
•Assumed use of blinds or drapes as in typical residences

Wall Fraction Lit: 0.90(10% of each wall shaded)
•Assumed, based on photographs of Fort Knox housing

TABLE B.4 (continued)

Roof Fraction Lit: 1.00 (no shading)

Door Crack Length: 30 ft.

Door Air Flow Coefficients: C: 40 N: 0.50
• See Table B.16

Window Crack Length: 296 ft.

Window Air Flow Coefficients: C: 1.7 N: 0.66
• See Table B.16

Wall Air Flow Coefficients: C: 0.01 N: 0.80
• See Table B.16

Peak Ventilation: 987 CFM
• Assumed one air change per hour as per Army measurements and typical residential data

Connected Electrical Load: 2.08 KW
• Primarily lighting at 0.50 watts/ft^2 total floor area

Peak Domestic Hot Water Demand: 6509 BTU/hr
• Assumed peak of 12 gal/hr for a family of four as per ASHRAE Systems, 1973 [1]

Winter Room Temperature: 68 °F (minimum)

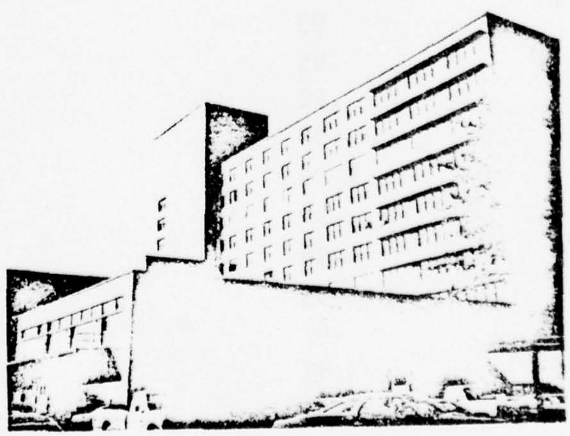
Summer Room Temperature: 75 °F (maximum)

TABLE B.4 (continued)

Time		Building Use Factor (for electrical equipment and ventilation)	Domestic Hot Water Use Factor (from Ref.1 for Apartments)
12		.81	.17
1		.67	.14
2		.61	.13
3		.58	.10
4	AM	.52	.11
5		.49	.10
6		.52	.10
7		.59	.13
8		.66	.15
9		.69	.25
10		.79	.12
11		.90	.19
12		.93	.17
1		.96	.18
2		.96	.15
3		.93	.13
4	PM	.95	.12
5		.93	.12
6		.98	.15
7		1.00	.19
8		.99	.21
9		.96	.18
10		.93	.15
11		.87	.13
12		.81	.17

Figure B.5
Type 5: Ft. Knox Large Hospital
Unit: Large Hospital

Building number 851



Building number 851

TABLE B.5

BUILDING INPUT DATA

Type 5: Ft. Knox Large Hospital

Unit: Large Hospital

Wall Area: 20,700 ft^2

Wall Composite Thermal Resistance: 4.0 $hr \cdot ft^2 \cdot °F/BTU$

Window Area: 13,200 ft^2

Window Thermal Resistance: 0.89 $hr \cdot ft^2 \cdot °F/BTU$
Assumed single pane, no storm windows

Roof Area: 8000 ft^2

Roof Composite Thermal Resistance: 16.0 $hr \cdot ft^2 \cdot °F/BTU$

Basement Ground-Contact Area: 11,000 ft^2
Assumed one underground level

Basement Wall Thermal Resistance: 6.5 $hr \cdot ft^2 \cdot °F/BTU$

Building Height: 81 ft

Building Orientation: 45° from North

Wall Surface Material: Concrete

Roof Surface Material: Asphalt

Wall Solar Absorptivity: 0.91

Roof Solar Absorptivity: 0.95

Window Shading Coefficient: 0.40 (40% of incident radiation transmitted)
Assumed use of shades as shown in hospital photographs

Wall Fraction Lit: 0.95 (5% of each wall shaded)
Assumed, based on photographs

Roof Fraction Lit: 1.00 (no shading)

Door Crack Length: 147 ft

Door Air Flow Coefficients: C: 40 N: 0.50
See Table B.16

TABLE B.5 (Continued)

Window Crack Length:	5,600 ft
Window Air Flow Coefficients:	C: 2.2 N: 0.66 See Table B.16 Based on non-weatherstripped window in masonry frame with caulking
Wall Air Flow Coefficients:	C: 0.004 N: 0.70 See Table B.16
Peak Ventilation:	17,000 CFM Assumed three air changes per hour as average of Army recommendations for various areas in a hospital ranging from one to twelve air changes per hour.
Connected Electrical Load:	31 KW Average of total demand given as .82 watts/ft^2 total floor area
Peak Domestic Hot Water Demand:	230,000 BTU/hr Assumed peak of 4.25 gal/hr per bed, 100 beds.
Winter Room Temperatures:	74°F (constant)
Summer Room Temperatures:	74°F (constant)

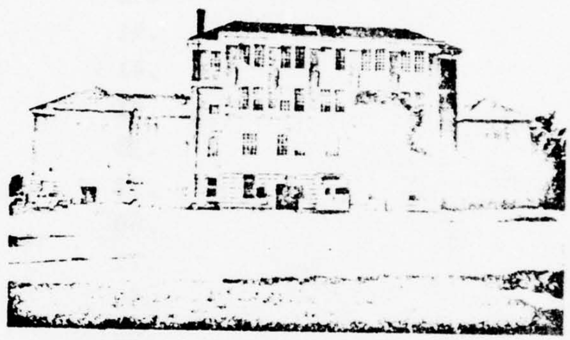
TABLE B.5 (Continued)

Time	Building Use Factor (for electrical equipment and ventilation)	Domestic Hot Water Use Factor (varies with building use)
12	.53	
1	.41	
2	.41	
3	.41	
4	.41	
5 AM	.38	
6	.53	
7	.60	
8	.71	
9	.88	
10	.94	
11	.98	
12	.99	
1	1.00	
2	1.00	
3	1.00	
4	.99	
5 PM	.93	
6	.79	
7	.70	
8	.70	
9	.68	
10	.59	
11	.56	
12	.53	

Figure B.6

Type 6: Ft. Knox Small Hospital

Unit: Small Hospital



Building number 1030

Building number 1030

TABLE B.6

BUILDING INPUT DATA

Type 6: Ft. Knox Small Hospital

Unit: Small Hospital

Wall Area: 13,200 ft^2

Wall Composite Thermal Resistance: 3.45 $hr \cdot ft^2 \cdot {}^\circ F/BTU$

Window Area: 2,775 ft^2

Window Thermal Resistance: 0.89 $hr \cdot ft^2 \cdot {}^\circ F/BTU$
Assumed single pane, no storm windows

Roof Area: 9,600 ft^2

Roof Composite Thermal Resistance: 20.0 $hr \cdot ft^2 \cdot {}^\circ F/BTU$

Basement Ground-Contact Area: 11,800 ft^2
Assumed one basement level

Basement Wall Thermal Resistance: 6.5 $hr \cdot ft^2 \cdot {}^\circ F/BTU$

Building Height: 30 ft

Building Orientation: 45° from North

Wall Surface Material: Brick

Roof Surface Material: Slate

Wall Solar Absorptivity: 0.91

Roof Solar Absorptivity: 0.80

Window Shading Coefficient: 0.40 (40% of incident radiation transmitted
Assumed use of shades as shown in hospital photograph

Wall Fraction Lit: 0.85 (15% of each wall shaded)
Assumed, based on photographs

Roof Fraction Lit: 1.00 (no shading)

TABLE B.6 (Continued)

Door Crack Length: 93 ft

Door Air Flow Coefficients: C: 40 N: 0.50
See Table B.16

Window Crack Length: 1,665 ft

Window Air Flow Coefficients: C: 3.2 N: 0.66
See Table B.16

Wall Air Flow Coefficients: C: 0.004 N: 0.7
See Table B.16

Peak Ventilation: 12,500 CFM
Assumed three air changes per hour as average of Army recommendations for various areas in a hospital ranging from one to twelve air changes per hour

Connected Electrical Load: 20 KW
Average of total demand given as .82 watts/ft^2 total floor area

Peak Domestic Hot Water Demand: 115,000 BTU/hr
Assumed peak of 4.25 gal/hr per bed, 50 beds

Winter Room Temperatures: 74°F (constant)

Summer Room Temperatures: 74°F (constant)

TABLE B.6 (Continued)

Time	Building Use Factor (for electrical equipment, ventilation and hot water usage)
12	.53
1	.41
2	.41
3	.41
4	.41
5 AM	.38
6	.53
7	.60
8	.71
9	.88
10	.94
11	.98
12	.99
1	1.00
2	1.00
3	1.00
4	.99
5	.93
6 PM	.79
7	.70
8	.70
9	.68
10	.59
11	.56
12	.53

TABLE B.7

BUILDING INPUT DATA

Type 7: Community

Unit: Recreation/Community Center

Wall Area: 10,556 ft^2

Wall Composite Thermal Resistance: 3.13(hr)(ft^2)(°F)/BTU

Window Area: 300 ft^2

Window Thermal Resistance: 0.89(hr)(ft^2)(°F)/BTU
·Assumed single pane, no storm windows

Roof Area: 20,369 ft^2

Roof Composite Thermal Resistance: 5.89(hr)(ft^2(°F)/BTU

Basement Ground-Contact Area: 20,486 ft^2

Basement Wall Thermal Resistance: 5.27(hr)(ft^2)(°F)/BTU

Building Height: 15 ft.

Building Orientation: 45° from North

Wall Surface Material: Concrete Block

Roof Surface Material: Asphalt Shingle

Wall Solar Absorptivity: 0.68

Roof Solar Absorptivity: 0.80

Window Shading Coefficient: 0.70(70% of incident radiation transmitted)
·Assumed, based upon photograph of community center

Wall Fraction Lit: 1.00 (no shading)

Roof Fraction Lit: 1.00 (no shading)

TABLE B.7 (continued).

Door Crack Length: 66 ft.

Door Air Flow Coefficients: C: 40 N: 0.50
•See Table B.16

Window Crack Length: 150 ft.

Window Air Flow Coefficients: C: 3.0 N: 0.66
•See Table B.16

Wall Air Flow Coefficients: C: 0.004 N: 0.80
•See Table B.16

Peak Ventilation: 25,624 CFM
•Assumed five air changer per hour as per Army measurements

Connected Electrical Load: 30.73 KW
•Primarily lighting at 1.5 watts/ ft^2 total floor area

Peak Domestic Hot Water Demand: Unavailable

Winter Room Temperatures: 68 °F (minimum)

Summer Room Temperatures: 78 °F (maximum)

TABLE B.7 (continued)

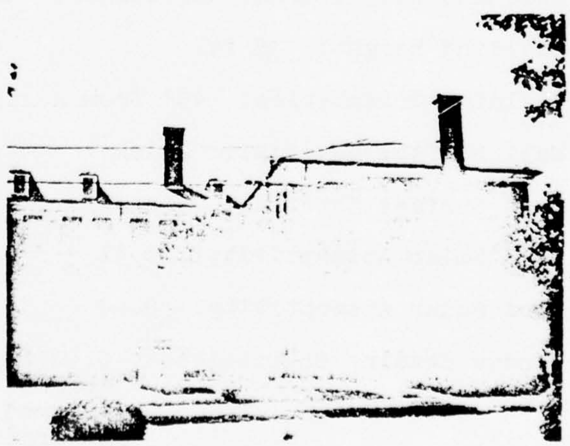
Time	Building Use Factor (for electrical equipment and ventilation)
12	.15
1	.15
2	.15
3	.15
4 AM	.15
5	.15
6	.15
7	.15
8	.15
9	.25
10	.50
11	1.00
12	1.00
1	1.00
2	1.00
3	1.00
4 PM	1.00
5	1.00
6	1.00
7	1.00
8	1.00
9	.90
10	.90
11	.80
12	.15

Figure B.8

Type 8: Administration and Training

Unit: Training Building

Building number 1468



Building number 1468

TABLE B.8

BUILDING INPUT DATA

Type 8: Administration and Training

Unit: Training Building

Wall Area: 14,782 ft^2

Wall Composite Thermal Resistance: 4.00(hr)(ft^2)(°F)/BTU

Window Area: 5666 ft^2

Window Thermal Resistance: 0.89(hr)(ft^2)°F)/BTU
•Assumed single pane, no storm windows

Roof Area: 8135 ft^2

Roof Composite Thermal Resistance: 20.00(hr)(ft^2)(°F)/BTU

Basement Ground-Contact Area: 8135 ft^2

Basement Wall Thermal Resistance: 4.04(hr)(ft^2)(°F)/BTU

Building Height: 35 ft.

Building Orientation: 45° from North

Wall Surface Material: Brick

Roof Surface Material: Slate

Wall Solar Absorptivity: 0.91

Roof Solar Absorptivity: 0.80

Window Shading Coefficient: 0.60(60% of incident radiation transmitted)
•Assumed use of shades as shown in training building photograph

Wall Fraction Lit: 0.90(10% of each wall shaded)
•Assumed, based on training building photograph

TABLE B.8 (continued)

Roof Fraction Lit: 1.00 (no shading)

Door Crack Length: 90 ft.

Door Air Flow Coefficients: C: 40 N: 0.50
•See Table B.16

Window Crack Length: 1304 ft.

Window Air Flow Coefficients: C: 3.2 N: 0.66
•See Table B.16

Wall Air Flow Coefficients: C: 0.004 N: 0.80
•See Table B.16

Peak Ventilation: 7813 CFM
•Assumed 2.5 air changes per hour as per Army measurements and typical office building data

Connected Electrical Load: 72.34 KW
•Primarily lighting at 3 watts/ ft^2 total floor area

Peak Domestic Hot Water Demand: Negligible

Winter Room Temperatures: 68 °F (minimum)

Summer Room Temperatures: 75 °F (maximum)

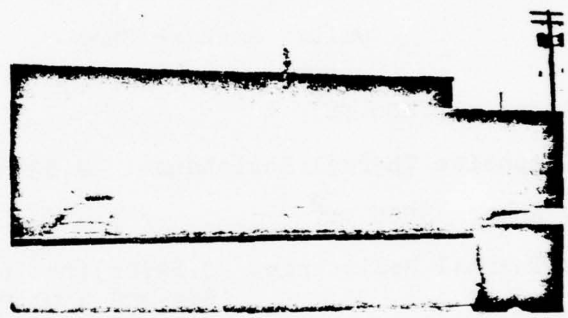
TABLE B.8 (continued)

Time		Building Use Factor (for electrical equipment and ventilation)
12		.05
1		.05
2		.05
3		.05
4	AM	.05
5		.05
6		.10
7		.30
8		.80
9		.95
10		.95
11		.95
12		.75
1		.75
2		.95
3		.95
4	PM	.50
5		.30
6		.10
7		.10
8		.10
9		.10
10		.05
11		.05
12		.05

Figure B.9

Type 9: Operations and Maintenance

Unit: Machine Shop



Building number 5935

Building number 5935

TABLE B.9

BUILDING INPUT DATA

Type 9: Operations and Maintenance

Unit: Machine Shop

Wall Area: 17,800 ft^2

Wall Composite Thermal Resistance: 2.63(hr)(ft^2)(°F)/BTU

Window Area: 3560 ft^2

Window Thermal Resistance: 0.89(hr)(ft^2)(°F)/BTU
·Assumed single pane, no storm windows

Roof Area: 41,850 ft^2

Roof Composite Thermal Resistance: 5.26(hr)(ft^2)(°F)/BTU

Basement Ground-Contact Area: 41,850 ft^2

Basement Wall Thermal Resistance: 4.00(hr)(ft^2)(°F)/BTU

Building Height: 20 ft.

Building Orientation: 45° from North

Wall Surface Material: Concrete Block/Brick

Roof Surface Material: Asphalt shingle

Wall Solar Absorptivity: 0.70

Roof Solar Absorptivity: 0.80

Window Shading Coefficient: 0.80(80% of incident radiation transmitted
·Assumed windows generally dirty and partly obstructed

Wall Fraction Lit: 1.00 (no shading)

Roof Fraction Lit: 1.00 (no shading)

TABLE B.9 (continued)

Door Crack Length: 158 ft.

Door Air Flow Coefficients: C: 40 N: 0.50
•See Table B.16

Window Crack Length: 3816 ft.

Window Air Flow Coefficients: C: 3.2 N: 0.66
•See Table B.16

Wall Air Flow Coefficients: C: 0.004 N: 0.80
•See Table B.16

Peak Ventilation: 13,958 CFM
•Assumed one air change per hour as recommended for light manufacturing facilities

Connected Electrical Load: 41.85 KW
•Primarily lighting at 1.0 watt/ft^2 total floor area

Peak Domestic Hot Water Demand: Negligible

Winter Room Temperature: 65 °F (minimum)

Summer Room Temperatures: Not air conditioned

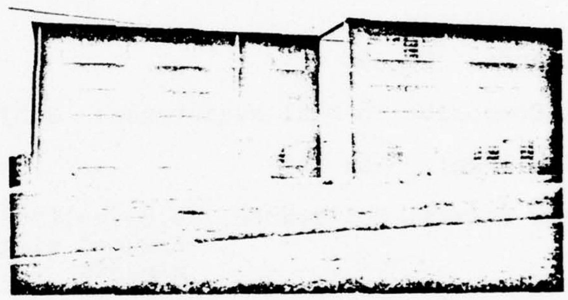
TABLE B.9 (continued)

Time		Building Use Factor (for electrical equipment and ventilation)
12		.15
1		.10
2		.10
3		.10
4	AM	.10
5		.10
6		.10
7		.10
8		.50
9		.75
10		.80
11		1.00
12		.95
1		.95
2		.90
3		.90
4	PM	.75
5		.75
6		.35
7		.15
8		.15
9		.15
10		.15
11		.15
12		.15

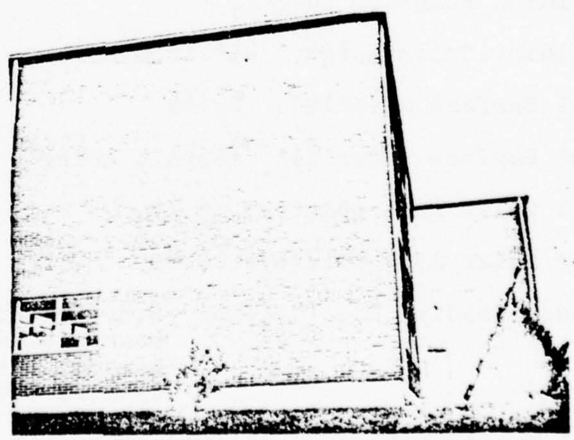
Figure B.10

Type 10: Troop Housing: Brick

Unit: Barracks



Building number 5937



Building number 5937

TABLE B.10

BUILDING INPUT DATA

Type 10: Troop Housing: Brick

Unit: Barracks Unit

Wall Area: 25,598 ft^2

Wall Composite Thermal Resistance: 2.63(hr)(ft^2)(°F)/BTU

Window Area: 5261 ft^2

Window Thermal Resistance: 0.89(hr)(ft^2)(°F)/BTU
- Assumed single pane, no storm windows

Roof Area: 18,685 ft^2

Roof Composite Thermal Resistance: 5.26(hr)(ft^2)(°F)/BTU

Basement Ground-Contact Area: 18,685 ft^2

Basement Wall Thermal Resistance: 4.00(hr)(ft^2)(°F)/BTU

Building Height: 28.5 ft

Building Orientation: 45° from North

Wall Surface Material: Brick

Roof Surface Material: Asphalt shingle

Wall Solar Absorptivity: 0.70

Roof Solar Absorptivity: 0.80

Window Shading Coefficient: 0.50(50% of incident radiation transmitted)
- Assumed use of shades as in typical residences

Wall Fraction Lit: 0.80 (20% of each wall shaded)
- Assumed, based on photographs of typical residences

TABLE B.10 (continued)

Roof Fraction Lit: 1.00 (no shading)

Door Crack Length: 84 ft.

Door Air Flow Coefficients: C: 40 N: 0.50
•See Table B.16

Window Crack Length: 2368 ft.

Window Air Flow Coefficients: C: 3.2 N: 0.66
•See Table B.16

Wall Air Flow Coefficients: C: 0.004 N: 0.80
•See Table B.16

Peak Ventilation: 13,397 CFM
•Assumed 1.5 air changes per hour as per Army measurements and typical residential data

Connected Electrical Load: 127.40 KW
•Primarily lighting at 2.5 watts/ft^2 total floor area

Peak Domestic Hot Water Demand: 430,100 BTU/hr
•Assumed peak of 3.8 gph per person, 200 residents, as per ASHRAE Systems, 1973[1]

Winter Room Temperatures: 68 °F (minimum)

Summer Room Temperatures: 78 °F (maximum)

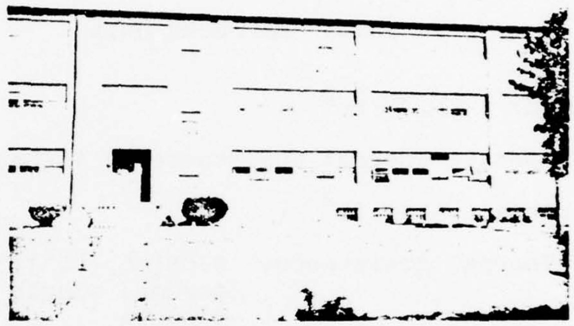
TABLE B.10 (continued)

Time		Building Use Factor (for electrical equipment and ventilation)	Domestic Hot Water Use Factor (from Ref.1 for Dormitories)
12		.20	.33
1		.20	.26
2		.20	.18
3		.20	.11
4	AM	.20	.03
5		.20	.04
6		.50	.01
7		.90	.11
8		.30	.18
9		.30	.21
10		.30	.22
11		.50	.18
12		.80	.24
1		.50	.16
2		.30	.13
3		.30	.16
4	PM	.50	.24
5		.70	.20
6		.80	.26
7		.80	.30
8		.80	.29
9		.80	.16
10		.50	.26
11		.20	.37
12		.20	.33

Figure B.11

Type 11: Troop Housing, Block

Unit: Barracks



Building number 2374

Building number 2374

TABLE B.11

BUILDING INPUT DATA

Type 11: Troop Housing: Block

Unit: Barracks Unit

Wall Area: 21,000 ft^2

Wall Composite Thermal Resistance: 2.44(hr)(ft^2)(°F)/BTU

Window Area: 5720 ft^2

Window Thermal Resistance: 0.89(hr)(ft^2)(°F)/BTU
- Assumed single pane, no storm windows

Roof Area: 17,200 ft^2

Roof Composite Thermal Resistance: 5.26(hr)(ft^2)(°F)/BTU

Basement Ground-Contact Area: 17,200 ft^2

Basement Wall Theraml Resistance: 4.50(hr)(ft^2)(°F)/BTU

Building Height: 30 ft.

Building Orientation: 45° from North

Wall Surface Materal: Concrete Block

Roof Surface Material: Asphalt Shingle

Wall Solar Absorptivity: 0.70

Roof Solar Absorptivity: 0.80

Window Shading Coefficient: 0.60 (60% of incident radiation transmitted)
- Assumed use of shades as in typical residences

Wall Fraction Lit: 1.00 (no shading)

Roof Fraction Lit: 1.00 (no shading)

TABLE B.11 (continued)

Door Crack Length: 88 ft.

Door Air Flow Coefficients: C: 40 N: 0.50
•See Table B.16

Window Crack Length: 2264 ft.

Window Air Flow Coefficients: C: 3.2 N: 0.66
•See Table B.16

Wall Air Flow Coefficients: C: 0.004 N: 0.80
•See Table B.16

Peak Ventilation: 12,818 CFM
•Assumed 1.5 air changes per hour as per Army measurements and typical residential data

Connected Electrical Load: 127.40 KW
•Primarily lighting at 2.5 watts/ ft^2 total floor area

Peak Domestic Hot Water Demand: 344,932 BTU/hr
•Assumed peak of 3.8 gph per person, 160 residents, as per ASHRAE Systems, 1973 [1]

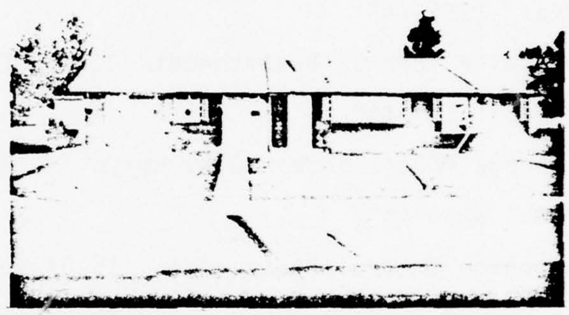
Winter Room Temperatures: 68 °F (minimum)

Summer Room Temperatures: 78 °F (maximum)

TABLE B.11 (continued)

Time		Building Use Factor (for electrical equipment and ventilation)	Domestic Hot Water Use Factor (from Ref.1 for Dormitories)
12		.20	.33
1		.20	.26
2		.20	.18
3		.20	.11
4	AM	.20	.10
5		.20	.10
6		.50	.10
7		.90	.11
8		.30	.18
9		.30	.21
10		.30	.22
11		.50	.18
12		.80	.24
1		.50	.16
2		.30	.13
3		.30	.16
4	PM	.50	.24
5		.70	.20
6		.80	.26
7		.80	.30
8		.80	.29
9		.80	.16
10		.50	.26
11		.20	.37
12		.20	.33

Figure B.12
Type 12: Family Housing
Unit: Wood Duplex



Building number 7528

TABLE B.12

BUILDING INPUT DATA

Type 12: Family Housing

Unit: Wood Duplex

Wall Area: 1356 ft^2

Wall Composite Thermal Resistance: 3.7 $hr \cdot ft^2 \cdot °F/BTU$

Window Area: 144 ft^2

Window Thermal Resistance: 0.89 $hr \cdot ft^2 \cdot °F/BTU$

Roof Area: 1250 ft^2

Roof Composite Thermal Resistance: 16.67 $hr \cdot ft^2 \cdot °F/BTU$

Basement Ground-Contact Area: 1250 ft^2

Basement Wall Thermal Resistance: 4.91 $hr \cdot ft^2 \cdot °F/BTU$

Building Height: 10 ft

Building Orientation: 45° from North

Wall Surface Material: Wood

Roof Surface Material: Asphalt Shingle

Wall Solar Absorptivity: .80

Roof Solar Absorptivity: .80

Window Shading Coefficient: .50

Wall Fraction Lit: .80

Roof Fraction Lit: 1.00

Door Crack Length: 20 ft

Door Air Flow Coefficients: C: 40 N: 0.50 (see Table B.16)

Window Crack Length: 40 ft

Window Air Flow Coefficients: C: 1.7 N: 0.66 (see Table B.16)

TABLE B.12 (Continued)

Wall Air Flow Coefficients: C: 1.2 N: 0.66 (see Table B.16)

Peak Ventilation: 208 CFM
(one air change per hour)

Connected Electrical Load: 0.91 KW
(lighting .87 watts/ft^2)

Peak Domestic Hot Water Demand: 6,009 BTU/hr (4 people)

Winter Room Temperature: 68°F (minimum)

Summer Room Temperature: 75°F (maximum)

TABLE B.12 (Continued)

Time	Building Use Factor (for electrical equipment and ventilation)	Domestic Hot Water Use Factor (from ASHRAE)
12	.81	.17
1	.67	.14
2	.61	.13
3	.58	.10
4	.52	.11
5 AM	.49	.10
6	.52	.10
7	.59	.13
8	.66	.15
9	.69	.25
10	.79	.21
11	.90	.19
12	.93	.17
1	.96	.18
2	.96	.15
3	.93	.13
4	.95	.12
5	.93	.12
6 PM	.98	.15
7	1.00	.19
8	.99	.21
9	.96	.18
10	.93	.15
11	.87	.13
12	.81	.17

Figure B.13
Type 13: Family Housing
Unit: Single Family

Building number 4267

Building number 4267

TABLE B.13

BUILDING INPUT DATA

Type 13: Family Housing

Unit: Single Family

Wall Area: 1420 ft^2

Wall Composite Thermal Resistance: 3.7 $hr \cdot ft^2 \cdot °F/BTU$

Window Area: 180 ft^2

Window Thermal Resistance: .89 $hr \cdot ft^2 \cdot °F/BTU$

Roof Area: 1200 ft^2

Roof Composite Thermal Resistance: 16.67 $hr \cdot ft^2 \cdot °F/BTU$

Basement Ground-Contact Area: 1200 ft^2

Basement Wall Thermal Resistance: 4.91 $hr \cdot ft^2 \cdot °F/BTU$

Building Height: 10 ft

Building Orientation: 45° from North

Wall Surface Material: Stucco

Roof Surface Material: Asphalt Shingle

Wall Solar Absorptivity: 0.50

Roof Solar Absorptivity: 0.80

Window Shading Coefficient: 0.50 (50% of incident radiation transmitted)
Assumed used of blinds, shades or drapes as per photographs

Wall Fraction Lit: 0.80 (20% of each wall shaded, as per photographs)

Roof Fraction Lit: 1.00 (no shading)

Door Crack Length: 21 ft

Door Air Flow Coefficients: C: 40 N: 0.50
See Table B.16

TABLE B.13 (Continued)

Window Crack Length: 104 ft

Window Air Flow Coefficients: C: 1.7 N: 0.66
See Table B.16

Wall Air Flow Coefficients: C: 0.004 N: 0.70

Peak Ventilation: 200 CFM
(Assumed one air change per hour per typical residential data)

Connected Electrical Load: 1.044 KW
Primarily lighting at .82 KW/ft^2 of total floor area

Peak Domestic Hot Water Demand: 6,509 BTU/hr
Assumed peak of 12 gal/hr to serve one family as per ASHRAE Systems, 1973

Winter Room Temperature: 68°F (minimum)

Summer Room Temperature: 75°F (maximum)

TABLE B.13 (Continued)

Time	Building Use Factor (for electrical equipment and ventilation	Domestic Hot Water Use Factor (Ref. ASHRAE)
12	.81	.17
1	.67	.14
2	.61	.13
3	.58	.10
4 AM	.52	.11
5	.49	.10
6	.52	.10
7	.59	.13
8	.66	.15
9	.69	.25
10	.79	.21
11	.90	.19
12	.93	.17
1	.96	.18
2	.96	.15
3	.93	.13
4 PM	.95	.12
5	.93	.12
6	.98	.15
7	1.00	.19
8	.99	.21
9	.96	.18
10	.93	.15
11	.87	.13
12	.81	.17

TABLE B.14

BUILDING INPUT DATA

Type 14: Storage

Unit: Warehouse

Wall Area: 7104 ft^2

Wall Composite Thermal Resistance: 3.70(hr)(ft^2)(°F)/BTU

Window Area: 1420 ft^2

Window Thermal Resistance: 0.89(hr)(ft^2)(°F)/BTU
- Assumed single pane, no storm windows

Roof Area: 11,421 ft^2

Roof Composite Thermal Resistance: 3.33(hr)(ft^2)(°F)/BTU

Basement Ground-Contact Area: 11,421 ft^2

Basement Wall Thermal Resistance: 6.00(hr)(ft^2)(°F)/BTU

Building Height: 15.75 ft.

Building Orientation: 45° from North

Wall Surface Material: Concrete Block

Roof Surface Material: Asphalt Shingle

Wall Solar Absorptivity: 0.68

Roof Solar Absorptivity: 0.80

Window Shading Coefficient: 0.80(80% of incident radiation transmitted)
- Assumed some windows dirty or blocked by stored goods

Wall Fraction Lit: 1.00 (no shading)

Roof Fraction Lit: 1.00 (no shading)

Door Crack Length: 176 ft.

TABLE B.14 (continued)

Door Air Flow Coefficients: C: 40 N: 0.50
·See Table B.16

Window Crack Length: 222 ft.

Window Air Flow Coefficients: C: 2.2 N: 0.66
·See Table B.16

Wall Air Flow Coefficients: C: 0.01 N: 0.80
·See Table B.16

Peak Ventilation: 3001 CFM
·Assumed one air change per hour as conservative requirement

Connected Electrical Load: 22.84 KW
·Primarily lighting at 2.0 watts/ft^2 total floor area as per Army specifications

Peak Domestic Hot Water Demand: Negligible

Winter Room Temperatures: 40 °F (minimum)

Summer Room Temperatures: Not air conditioned

TABLE B.14 (continued)

Time		Building Use Factor (for electrical equipment and ventilation)
12		.10
1		.10
2		.10
3		.10
4	AM	.10
5		.10
6		.10
7		.30
8		.30
9		.95
10		.95
11		.95
12		.95
1		.95
2		.95
3		.95
4	PM	.95
5		.50
6		.10
7		.10
8		.10
9		.10
10		.10
11		.10
12		.10

TABLE B.15

BUILDING INPUT DATA

Type 15: Family Housing

Unit: Multiplex

Wall Area: 5220 ft^2

Wall Composite Thermal Resistance: 3.4 hr·ft^2·°F/BTU

Window Area: 2600 ft^2

Window Thermal Resistance: .89 hr·ft^2·°F/BTU
Single pane, no storm windows

Roof Area: 6000 ft^2

Roof Composite Thermal Resistance: 16.67 hr·ft^2·°F/BTU

Basement Ground-Contact Area: 6000 ft^2

Basement Wall Thermal Resistance: 4.91 hr·ft^2·°F/BTU

Building Height: 17 ft

Building Orientation: 45° from North

Wall Surface Material: Brick

Roof Surface Material: Asphalt Shingle

Wall Solar Absorptivity: .7

Roof Solar Absorptivity: .8

Window Shading Coefficient: .5

Wall Fraction Lit: .9

Roof Fraction Lit: 1.0

Door Crack Length: 240 ft

Door Air Flow Coefficient: C: 40 N: .5 (see Table B.16)

Window Crack Length: 520 ft

Window Air Flow Coefficients: C: 1.7 N: .66 (see Table B.16)

TABLE B.15 (Continued)

Wall Air Flow Coefficients: C: 0.01 N: .8 (See Table B.16)

Peak Ventilation: 1200 CFM

Connected Electrical Load: 7.32 KW

Peak Domestic Hot Water Demand: 42,000 BTU/hr

Winter Room Temperature: 68°F (minimum)

Summer Room Temperature: 75°F (maximum)

TABLE B.15 (Continued)

Time	Building Use Factor (for electrical equipment and ventilation	Domestic Hot Water Use Factor (ref. ASHRAE)
12	.81	.17
1 AM	.67	.14
2	.61	.13
3	.58	.10
4	.52	.11
5	.49	.10
6	.52	.10
7	.59	.13
8	.66	.15
9	.69	.25
10	.79	.21
11	.90	.19
12	.93	.17
1	.96	.18
2	.96	.15
3	.93	.13
4	.95	.12
5 PM	.93	.12
6	.98	.15
7	1.00	.19
8	.99	.21
9	.96	.18
10	.93	.15
11	.87	.13
12	.81	.17

TABLE B.16

INFILTRATION AIR FLOW COEFFICIENTS

(from Table A.17, Ref. 3)

Note: These coefficients are used to determine the infiltration air flow rates through:

$$I = C\Delta P^N$$

where I = infiltration, CFM per linear crack foot (or per square foot of wall area)**
ΔP = pressure difference across opening, in inches of water

		C	N
1.	Double-hung windows (locked)*		
	non-weatherstripped, loose fit	6	0.66
	average fit	2	0.66
	weatherstripped, loose fit	2	0.66
	average fit	1	0.66
2.	Window frames*		
	masonry frame with no caulking	1.2	0.66
	masonry frame with caulking	0.2	0.66
	wooden frame	1.0	0.66
3.	Swinging doors*		
	1/2" crack	160	0.5
	1/4" crack	80	0.5
	1/8" crack	40	0.5
4.	Walls**		
	8" plain brick	1	0.8
	8" brick and plaster	0.01	0.8
	13" plain brick	0.8	0.8
	13" brick and plaster	0.004	0.7
	13" brick, furring, lath and plaster	0.03	0.9
	frame wall, lath and plaster	0.01	0.55
	24" shingles on 1x6 boards on 14" centers	9	0.66
	16" shingles on 1x4 boards on 5" centers	5	0.66
	24" shingles on shiplap	3.6	0.7
	16" shingles on shiplap	1.2	0.66

*Values of C listed for these openings are per ft. of linear crack length.

**Values of C listed for the walls are per unit area of the wall surface.

REFERENCES

1. J. W. Stetkar and M. W. Golay, User's Manual: TDIST, A Program for Community Energy Demand Analysis and Total Energy System Response Simulation, FESA-RT-2021, MIT Nuclear Engineering Dept. (1976).

2. W. E. Evers, "E-CUBE Computerized Energy Analysis: Energy-Equipment-Economics," ASHRAE Journal, (Sept. 1971).

3. T. Kusuda, "NBSLD, Heating and Cooling Load Calculation Program," National Bureau Standards (1974).

4. J. P. Barnett and S. T. Liu, "A Comparative Study of Building Energy Analysis for a Multi-Family High-Rise Apartment Building in Omaha, Nebraska," National Bureau of Standards (1976).

APPENDIX C.1

HEAT EXCHANGER DESIGN

A wide variety of heat exchanger sizes and designs are commercially available to achieve a given desired heat transfer rating. The ultimate heat exchanger choice for a given application will depend upon the size and design of the associated piping system and upon energy consumption and capital cost criteria established by the designer. Tube fins and vanes and multi-pass flow geometries are commonly used to increase effectively the heat transfer area of a unit without greatly altering its physical dimensions. Tradeoffs among the number of tubes, their lengths and their diameters also affect the total heat transfer area and the fluid pressure losses in the exchanger for a given fluid mass flowrate. Because of its design simplicity and ease of analysis in calculating flowstream and energy transfer effects, a single-pass, counterflow, straight tube heat exchanger geometry was assumed to be used in all of the utility system simulations. The final system design and optimization criteria will not be significantly affected by the actual heat exchanger geometries chosen as long as the specified heat transfer ratings are met and the pressure losses through the units do not greatly exceed those calculated for the straight tube models used in sizing the distribution loop piping and the circula-

tion pumps.

The Tubular Exchanger Manufacturers Association (TEMA) has established a set of design and construction standards to be met by commercially available heat exchangers. [1] Table C.1 lists their preferred tube gages for Class C heat exchangers designed for commercial and general process applications. The standard tube bundle patterns are triangular and square matrices, and pressure ratings vary from 150 to 2500 psig. [1] All heat exchangers for the proposed thermal utility system were assumed to have 1" O.D., 14 gage steel tubes in square bundles with 1" diameter interstitial flow channels. The tube diameters were chosen to provide "reasonable" physical dimensions for the heat exchangers within the geometrical constraints of the computer models, but no additional optimization of the exchanger sizes was performed. The same tubes were assumed to be used in each unit to provide uniform design criteria for all the heat exchangers. The heat transfer coefficient for a tube in one of these heat exchangers can be calculated through:

$$U = \frac{1}{\frac{1}{h_o} + r_o + r_w + r_i\left(\frac{A_o}{A_i}\right) + \frac{1}{h_i}\left(\frac{A_o}{A_i}\right)} \qquad (C.1)$$

where U = heat transfer coefficient in BTU/hrft^2°F referred to the tube outer surface,

TABLE C.1

TEMA PREFERRED TUBE GAGES FOR CLASS.C HEAT EXCHANGERS

(from Ref. 1)

Tube O.D., inches	BWG	Wall Thickness, inches	Material
1/4	24	.022	Copper
3/8	22	.028	Copper
1/2	20	.035	Copper
5/8	18	.049	Copper
3/4	16	.065	Copper
	14	.083	Steel
	18	.049	Alloy
1	16	.065	Copper
	14	.083	Steel
1-1/4	14	.083	Steel

h_o = film coefficient of fluid outside tube,

h_i = film coefficient of fluid inside tube,

r_o = fouling resistance of outer surface of tube,

r_i = fouling resistance of inner surface of tube,

r_w = resistance of tube wall referred to outer surface,

A_o = tube outer surface area, and

A_i = tube inner surface area.

According to El-Wakil [2], the film heat transfer coefficient for water can be approximated by:

$$h = 0.00134(T + 100)\,\frac{V^{0.8}}{D_c^{0.2}} \tag{C.2}$$

where h = film heat transfer coefficient in BTU/hrft^2°F,

T = bulk fluid temperature or mean film temperature if temperature drop across film is $\geq$ 10 °F,

V = fluid velocity in ft/hr, and

D_c = channel diameter in ft.

The thermal resistance of the tube wall referred to its outer surface is given by TEMA as:

$$r_w = \frac{t_w}{12k_w}\left(\frac{d}{d - t_w}\right) \tag{C.3}$$

where r_w = wall thermal resistance in $(BTU/hrft^2{}^\circ F)^{-1}$,

t_w = tube wall thickness in inches,

k_w = tube wall thermal conductivity in BTU/hrft°F, and

d = tube O.D. in inches.

Finally, the fouling resistance measured for a wide variety of water types and flow velocities is approximately 0.002 $(BTU/hrft^2{}^\circ F)^{-1}$ for water temperatures above 125 °F. [1]

If an average water temperature of 300 °F inside the tubes, a water temperature of 200 °F in the channels and fluid velocities of 3 feet/second are assumed, the heat transfer coefficient between a 1" O.D., 14 gage carbon steel tube and a 1" I.D. interstitial channel is easily calculated.

1. Tube outer surface film coefficient:

$$h = 0.00134(200 + 100)\left[\frac{(3 \times 3600)^{0.8}}{(\frac{1}{12})^{0.2}}\right]$$

$$= 1113.78 \text{ BTU/hrft}^2{}^\circ\text{F}$$

2. Tube inner surface film coefficient:

$$h = 0.00134(300 + 100)\left[\frac{(3 \times 3600)^{0.8}}{(\frac{.834}{12})^{0.2}}\right]$$

$$= 1539.95 \text{ BTU/hrft}^2{}^\circ\text{F}$$

3. Tube wall thermal resistance:

(thermal conductivity of carbon steel = 29 BTU/hrft°F)

$$r_w = \frac{.083}{(12)(29)}\left[\frac{1}{1 - .083}\right]$$

$$= 0.00026\ (\mathrm{BTU/hrft^2{}^\circ F})^{-1}$$

4. Outer/inner surface area ratio:

$$\frac{A_o}{A_i} = \frac{\pi d_o L}{\pi d_i L} = \frac{d_o}{d_i} = \frac{1}{.834} = 1.199$$

5. Fouling resistances:

$$r_o = r_i = 0.002\ (\mathrm{BTU/hrft^2{}^\circ F})^{-1}$$

6. Heat transfer coefficient:

$$U = \frac{1}{\frac{1}{1113.78} + .002 + .00026 + .002(1.199) + \frac{1}{1539.95}(1.199)}$$

$$= 157.87\ \mathrm{BTU/hrft^2{}^\circ F}.$$

Using this value of heat transfer coefficient, the design temperatures and maximum heat exchanger loads, the required heat transfer areas are calculated. Tables C.2-C.10 list heat exchanger identification numbers, heat transfer areas and costs.

Table C.12 lists load center identification numbers and the building type distribution applied to those load centers. To find the number and types of buildings applied to a given load center, one must first locate the number of the load center (in the lefthand column), then read across (under the building-type columns) to the number of buildings of that type applied to that load center. Usually, a given load center will have only two or three different building types applied to it, since actual building distribution on the base tends to be clustered according to function. Broadly speaking, the distribution for the entire base consists of an inner core of administration and training buildings, surrounded by troop and family housing.

In Table C.12, load centers are grouped by blocks according to thermal/electric split value. Individual load centers are deleted from the TUS as the split value is reduced. Table C.13 lists the load center blocks from Table C.12 which are used to construct a given TUS. For example, Table C.13 shows that a 60% split value TUS is made up of load center blocks A, B, C1, C2 and C3. The 0% TUS shows no load centers because the 0% TUS assumes all loads are supplied electrically and therefore no TUS exists.

Pipes, heat exchangers and load centers are identified by a four-digit code number. Figure C.1 shows the supply

FIGURE C.1

SCHEMATIC DIAGRAM OF FORT KNOX SECONDARY LOOP NUMBER 7

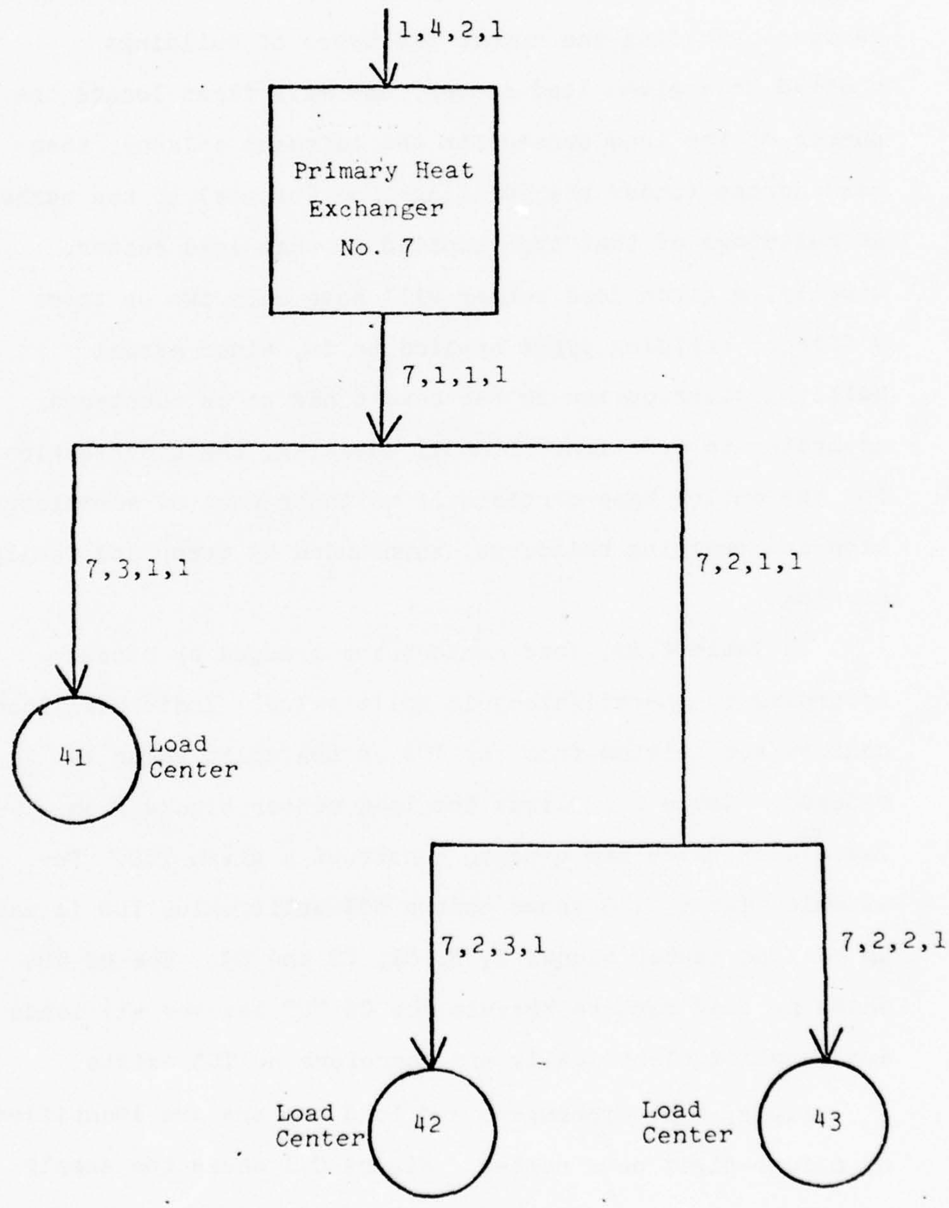


TABLE C.2

100% TUS HEAT EXCHANGER DESIGN AREAS AND COSTS

Option 1

Heat Exchanger Number	Peak Winter Heat Exchanger Area (Units of 1000 ft^2)	Cost (Units of \$100,000 in 1985)
1	3.53	5.27
2	1.52	5.15
3	2.51	5.21
4	1.90	5.06
5	1.92	5.07
6	2.21	5.19
7	.949	5.12
8	.961	5.12
9	2.41	5.20
10	2.46	5.21
13	2.97	5.24
14	1.08	5.12
	Total Cost	61.96

TABLE C.3

80% TUS HEAT EXCHANGER DESIGN AREAS AND COSTS

Option 1

Heat Exchanger Number	Peak Winter Heat Exchanger Area (Units of 1000 ft^2)	Cost (Units of $100,000 in 1985)
1	3.53	5.27
2	1.52	5.14
3	2.51	5.21
4	1.90	5.17
5	.192	5.07
6	2.21	5.19
7	.950	5.12
8	.962	5.12
9	2.41	5.20
10	2.46	5.21
	Total Cost	51.7

TABLE C.4

60% TUS HEAT EXCHANGER DESIGN AREAS AND COSTS

Option 1

Heat Exchanger Number	Peak Winter Heat Exchanger Area (Units of 1000 ft^2)	Cost (Units of $100,000 in 1985)
1	3.53	5.27
2	1.53	5.15
3	2.51	5.21
4	1.90	5.17
5	.192	5.07
6	2.21	5.14
7	.952	5.12
10	2.47	5.21
	Total Cost	41.4

TABLE C.5

40% TUS HEAT EXCHANGER DESIGN AREAS AND COSTS

Option 1

Heat Exchanger Number	Peak Winter Heat Exchanger Area (Units of 1000 ft^2)	Cost (Units of \$100,000 in 1985)
1	3.54	5.27
2	1.53	5.15
3	2.52	5.21
4	1.90	5.17
	Total Cost	20.8

TABLE C.6

20% TUS HEAT EXCHANGER DESIGN AREAS AND COSTS

Option 1

Heat Exchanger Number	Peak Winter Heat Exchanger Area (Units of 1000 ft^2)	Cost (Units of $100,000 in 1985)
2	1.53	5.15
3	2.52	5.09
4	1.91	5.17
	Total Cost	15.4

TABLE C.7

100% TUS HEAT EXCHANGER DESIGN AREAS AND COSTS

Option 2

Heat Exchanger Number	Peak Winter Heat Exchanger Area (Units of 1000 ft^2)	Cost (Units of $100,000 in 1985)
1	3.6	5.27
2	1.52	5.15
3	2.57	5.21
4	1.99	5.18
5	.192	5.07
6	2.32	5.20
7	.990	5.12
8	.972	5.12
9	2.41	5.21
10	2.49	5.21
13	3.03	5.24
14	1.10	5.12
	Total Cost	62.1

TABLE C.8

80% TUS HEAT EXCHANGER DESIGN AREAS AND COSTS

Option 2

Heat Exchanger Number	Peak Winter Heat Exchanger Area (Units of 1000 ft^2)	Cost (Units of $100,000 in 1985)
1	3.54	5.27
2	1.52	5.15
3	2.45	5.20
4	2.0	5.18
5	.192	5.07
6	2.31	5.20
7	.983	5.12
8	.95	5.12
9	2.36	5.20
10	2.48	5.21
	Total Cost	51.7

TABLE C.9

60% TUS HEAT EXCHANGER DESIGN AREAS AND COSTS

Option 2

Heat Exchanger Number	Peak Winter Heat Exchanger Area (Units of 1000 ft^2)	Cost (Units of $100,000 in 1985)
1	3.55	5.27
2	1.53	5.15
3	2.45	5.20
4	1.99	5.18
5	.192	5.07
6	2.31	5.20
7	.985	5.12
10	2.48	5.21
	Total Cost	41.4

TABLE C.10

40% TUS HEAT EXCHANGER DESIGN AREAS AND COSTS

Option 2

Heat Exchanger Number	Peak Winter Heat Exchanger Area (Units of 1000 ft^2)	Cost (Units of $100,000 in 1985)
1	3.62	5.27
2	1.53	5.15
3	2.57	5.21
4	2.00	5.18
	Total Cost	20.8

TABLE C.11

20% TUS HEAT EXCHANGER DESIGN AREAS AND COSTS

Option 2

Heat Exchanger Number	Peak Winter Heat Exchanger Area (Units of 1000 ft^2)	Cost (Units of \$100,000 in 1985)
2	1.53	5.15
3	2.58	5.21
4	2.00	5.18
	Total Cost	15.5

TABLE C.12

FT. KNOX BUILDING DISTRIBUTION

BUILDING DISTRIBUTION BY LOAD CENTER NUMBER

	TYPE / Load Center Number	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
	11	0.	0.	0.	0.	1.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
C1	12	0.	0.	0.	0.	0.	1.	0.	0.	0.	0.	0.	0.	0.	0.	0.
	15	0.	123.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
	16	0.	141.	4.	0.	0.	0.	0.	4.	0.	0.	0.	0.	0.	0.	0.
B	17	0.	0.	78.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
	18	0.	0.	64.	0.	0.	0.	1.	0.	0.	0.	0.	0.	0.	0.	0.
	19	0.	0.	0.	0.	0.	0.	0.	0.	6.	0.	0.	0.	0.	3.	0.
	20	0.	0.	0.	0.	0.	0.	0.	0.	4.	0.	0.	0.	0.	0.	0.
	21	0.	0.	0.	0.	0.	0.	0.	0.	1.	0.	0.	0.	0.	1.	0.
	22	0.	0.	0.	0.	0.	0.	0.	0.	3.	0.	0.	0.	0.	1.	0.
	23	0.	0.	0.	0.	0.	0.	0.	0.	9.	0.	0.	0.	0.	0.	0.
	24	0.	0.	0.	0.	0.	0.	2.	1.	0.	0.	10.	0.	0.	0.	0.
	25	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	2.	0.	0.	0.	0.
A	26	0.	0.	9.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
	27	0.	0.	0.	0.	0.	0.	0.	21.	0.	0.	0.	0.	0.	0.	0.
	28	0.	0.	0.	0.	0.	0.	0.	1.	3.	0.	0.	0.	0.	0.	0.
	29	0.	0.	0.	0.	0.	0.	0.	0.	6.	0.	0.	0.	0.	0.	0.
	30	0.	0.	0.	0.	0.	0.	4.	1.	0.	7.	0.	0.	0.	0.	0.
	31	0.	0.	0.	0.	0.	0.	1.	1.	1.	5.	0.	0.	0.	0.	0.
	32	0.	0.	0.	0.	0.	0.	1.	0.	0.	4.	0.	0.	0.	0.	0.
	33	0.	0.	0.	0.	0.	0.	1.	1.	0.	0.	0.	0.	0.	0.	0.
	34	0.	0.	0.	0.	0.	0.	0.	0.	2.	0.	0.	0.	0.	0.	0.
	35	0.	0.	0.	0.	0.	0.	0.	0.	1.	0.	0.	0.	0.	0.	0.
	36	0.	0.	0.	0.	0.	0.	0.	0.	2.	0.	0.	0.	0.	0.	0.
C2	37	0.	0.	0.	0.	0.	0.	1.	0.	0.	0.	9.	0.	0.	0.	0.
	38	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	8.	0.	0.	0.	0.
	39	0.	0.	0.	0.	0.	0.	1.	0.	0.	0.	5.	0.	0.	0.	0.
	40	0.	0.	0.	0.	0.	0.	1.	0.	0.	0.	6.	0.	0.	0.	0.
	41	0.	0.	0.	0.	0.	0.	0.	4.	0.	0.	6.	0.	0.	0.	0.
	42	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	3.	0.	0.	0.	0.
	43	0.	0.	0.	0.	0.	0.	0.	3.	0.	0.	2.	0.	0.	0.	0.

TABLE C.12 (Continued)

Type	Load Center Number	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
D	44	0.	6.	0.	0.	0.	0.	0.	0.	0.	0.	0.	9.	0.	0.	0.
	45	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	9.	0.	0.	0.
	46	0.	66.	1.	0.	0.	0.	1.	1.	0.	0.	0.	0.	0.	0.	0.
	47	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	20.	0.
	48	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	31.	0.	0.
	49	0.	0.	0.	0.	0.	0.	0.	2.	0.	0.	0.	0.	0.	0.	0.
	50	0.	0.	0.	0.	0.	0.	2.	0.	0.	0.	0.	0.	0.	0.	0.
	51	32.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	8.
	52	34.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	12.
	53	33.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	17.
	54	0.	24.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
	55	31.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	9.
	56	22.	0.	0.	0.	0.	0.	1.	0.	0.	3.	0.	0.	0.	0.	4.
C3	57	0.	0.	0.	23.	0.	0.	0.	0.	0.	0.	2.	0.	0.	0.	0.
	58	0.	0.	0.	11.	0.	0.	2.	0.	0.	0.	0.	0.	0.	0.	0.
	59	0.	0.	0.	18.	0.	0.	1.	1.	0.	0.	0.	0.	0.	0.	0.
	60	0.	0.	9.	56.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
	61	0.	0.	0.	14.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
	62	0.	0.	0.	30.	0.	0.	1.	0.	0.	0.	0.	0.	0.	0.	0.
	63	0.	0.	0.	0.	0.	0.	1.	1.	0.	0.	1.	0.	0.	6.	0.
E	64	0.	0.	0.	0.	0.	0.	0.	7.	0.	2.	0.	0.	0.	6.	0.
	65	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	3.	0.	0.	0.	0.
	66	0.	0.	33.	0.	0.	0.	11.	0.	0.	0.	0.	0.	0.	0.	0.
	67	0.	0.	0.	0.	0.	0.	1.	1.	0.	0.	0.	0.	0.	0.	0.
	68	0.	0.	50.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
	69	13.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
	70	28.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
	71	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	31.	0.	0.	0.
	72	46.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
	73	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	52.	0.	0.	0.
	74	79.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
***** COMMUNITY TOTALS		318.	360.	248.	152.	1.	1.	34.	50.	38.	21.	57.	101.	31.	37.	50.

TABLE C.13

LOAD CENTER DISTRIBUTION AS A FUNCTION OF

THERMAL/ELECTRIC SPLIT VALUE

Thermal/Electric Split Value	100%	80%	60%	40%	20%	0%
Load Center Block						
A	X	X	X	X	X	
B	X	X	X	X		
C1	X	X	X			
C2	X	X	X			
C3	X	X	X			
D	X	X				
E	X					

piping for Fort Knox secondary loop number 7. Primary heat exchanger number 7 receives 380°F water from the primary distribution system pipe number (1,4,2,1). Heat exchanger 7 supplies 200°F water to pipe (7,1,1,1), which carries water to a branch point supplying pipes (7,2,1,1) and (7,3,1,1). Pipe (7,3,1,1) supplies water to load center number 41. Pipe (7,2,1,1) supplies water to a branch point supplying pipes (7,2,2,1) and (7,2,3,1). These pipes supply hot water to load centers 43 and 42 respectively. For a complete discussion of the TDIST2 numbering conventions, refer to the TDIST2 User's Manual [3]. Briefly, each component in the TUS simulation is located in accordance with its four-number code. The first number in the group identifies the loop in which the component operates, the second, third and fourth digits locate the pipe in the loop. Table C.14 lists heat exchanger and load center numbers together with the pipes to which they are attached in the TUS. In addition to the four-number code, primary-to-secondary heat exchangers are listed with the number of the secondary loop which they supply. For example, heat exchanger number 1, located in pipe (1,2,3,1) supplies hot water to secondary loop number 13. Tables C.15 - C.19 list pipe identification codes, pipe lengths and cross-sectional areas for each pipe for each TUS option. Tables C.2 through C.19 are a complete record of all pipe and heat exchanger data actually used in the base's TUS simulations.

TABLE C.14

HEAT EXCHANGER AND LOAD CENTER

NODE IDENTIFICATION SYSTEM

Component Number	I	J	K	L	Loop
1	1	2	3	1	13
2	1	2	2	1	2
3	1	2	4	1	3
4	1	3	1	1	4
5	1	5	2	1	5
6	1	5	3	1	6
7	1	4	2	1	7
8	1	4	3	2	8
9	1	4	3	3	9
10	1	4	5	2	10
11	1	4	5	3	0
12	1	4	5	4	0
13	1	4	4	2	11
14	1	4	4	3	12
15	13	2	1	1	0
16	13	3	2	1	0
17	13	3	3	2	0
18	13	3	2	3	0
19	2	4	1	1	0
20	2	2	1	1	0
21	2	3	3	1	0
22	2	3	2	3	0
23	2	3	2	2	0
24	3	4	1	1	0
25	3	2	3	1	0
26	3	2	2	1	0
27	3	3	3	1	0
28	3	3	2	3	0
29	3	3	2	2	0
30	4	3	3	1	0
31	4	3	2	1	0
32	4	2	3	1	0
33	4	2	2	1	0
34	5	3	1	1	0

TABLE C.14 (Continued)

HEAT EXCHANGER AND LOAD CENTER

NODE IDENTIFICATION SYSTEM

Component Number	I	J	K	L	Loop
35	5	2	1	1	0
36	6	3	1	1	0
37	6	2	3	3	0
38	6	2	3	2	0
39	6	2	2	3	0
40	6	2	2	2	0
41	7	3	1	1	0
42	7	2	3	1	0
43	7	2	2	1	0
44	8	3	3	1	0
45	8	3	2	1	0
46	8	2	3	3	0
47	8	2	3	2	0
48	8	2	2	1	0
49	9	3	3	3	0
50	9	3	3	2	0
51	9	3	2	3	0
52	9	3	2	2	0
53	9	2	3	3	0
54	9	2	3	2	0
55	9	2	2	3	0
56	9	2	2	2	0
57	10	5	1	1	0
58	10	4	3	1	0
59	10	4	2	1	0
60	10	2	1	1	0
61	10	3	3	3	0
62	10	3	3	2	0
63	10	3	2	1	0
64	11	5	1	1	0
65	11	4	1	1	0
66	11	2	1	1	0
67	11	3	4	1	0
68	11	3	2	1	0
69	11	3	3	3	0
70	11	3	3	2	0
71	12	3	3	1	0
72	12	3	2	3	0
73	12	2	1	1	0
74	12	3	2	2	0

TABLE C.15

100% TUS PIPE DATA

I	J	K	L	PIPE LENGTH	PIPE AREA (1)	PIPE AREA (2)
1	1	1	1	16000.	1.39600	1.39600
1	2	2	1	400.	0.08840	0.08840
1	2	4	1	800.	0.20060	0.20060
1	4	1	1	8200.	0.54750	0.54750
1	4	3	1	3600.	0.20060	0.20060
1	4	3	3	3200.	0.20060	0.20060
1	4	4	2	800.	0.20060	0.20060
1	4	5	1	1200.	0.20060	0.20060
1	4	5	3	640.	0.00600	0.00600
1	5	1	1	2320.	0.20060	0.20060
1	5	3	1	2000.	0.20060	0.20060
2	2	1	1	1200.	0.34740	0.02330
2	3	2	1	1440.	1.39600	0.08840
2	3	2	3	50.	0.34740	0.02330
2	4	1	1	2400.	0.54750	0.05130
3	2	1	1	1000.	0.34740	0.02330
3	2	3	1	720.	0.20060	0.02330
3	3	2	1	1680.	0.77730	0.08840
3	3	2	3	960.	0.34740	0.02330
3	4	1	1	3440.	0.93970	0.08840
4	2	1	1	320.	0.54750	0.05130
4	2	3	1	480.	0.34740	0.05130
4	3	2	1	1760.	0.54750	0.05130
5	1	1	1	50.	0.34740	0.02330
5	3	1	1	800.	0.20060	0.02330
6	2	1	1	160.	2.64000	0.20060
6	2	2	2	880.	0.54750	0.05130
6	2	3	1	400.	1.39600	0.20060
6	2	3	3	1600.	0.77730	0.08840
7	1	1	1	320.	1.39600	0.20060
7	2	2	1	800.	0.34740	0.02330
7	3	1	1	1200.	0.77730	0.05130
8	2	1	1	960.	0.54750	0.08840
8	2	3	1	2240.	0.54750	0.08840
8	2	3	3	400.	0.54750	0.08840
8	3	2	1	800.	0.02330	0.00600
9	1	1	1	480.	1.76700	0.34740
9	2	2	1	240.	0.77730	0.08840
9	2	2	3	1200.	0.34740	0.05130
9	2	3	2	1280.	0.20060	0.02330
9	3	1	1	400.	0.77730	0.20060
9	3	2	2	1120.	0.34740	0.05130
9	3	3	1	560.	0.34740	0.05130
9	3	3	3	320.	0.20060	0.02330
10	2	1	1	1200.	0.54750	0.08840
10	3	2	1	160.	0.20060	0.02330
10	3	3	2	1200.	0.34740	0.05130
10	4	1	1	1120.	0.54750	0.08840
10	4	3	1	1520.	0.20060	0.05130

Notes:

(1) Pipes sized for absorptive air conditioning
(2) Pipes sized for winter peak load

TABLE C.15 (Continued)

11	1	1	1	50.	2.64000	0.34740
11	3	1	1	1200.	0.77730	0.08840
11	3	3	1	1200.	0.20060	0.05130
11	3	3	3	400.	0.08840	0.02330
11	4	1	1	800.	0.34740	0.02330
12	1	1	1	1600.	0.77730	0.20060
12	3	1	1	2800.	0.77730	0.08840
12	3	2	2	2400.	0.54750	0.05130
12	3	3	1	50.	0.08840	0.02330
13	2	1	1	480.	0.77730	0.08840
13	3	2	1	480.	0.93970	0.20060
13	3	3	2	50.	0.77730	0.08840
1	2	1	1	4640.	0.54750	0.54750
1	2	3	1	8320.	0.20060	0.20060
1	3	1	1	50.	0.20060	0.20060
1	4	2	1	80.	0.05130	0.05130
1	4	3	2	80.	0.05130	0.05130
1	4	4	1	6000.	0.20060	0.20060
1	4	4	3	3120.	0.08840	0.08840
1	4	5	2	80.	0.20060	0.20060
1	4	5	4	2320.	0.00600	0.00600
1	5	2	1	720.	0.02330	0.02330
2	1	1	1	50.	2.18200	0.20060
2	3	1	1	960.	1.39600	0.08840
2	3	2	2	800.	0.77730	0.08840
2	3	3	1	50.	0.08840	0.02330
3	1	1	1	400.	3.14200	0.34740
3	2	2	1	50.	0.08840	0.02330
3	3	1	1	1200.	2.18200	0.20060
3	3	2	2	560.	0.54750	0.05130
3	3	3	1	800.	1.39600	0.08840
4	1	1	1	50.	1.76700	0.20060
4	2	2	1	1440.	0.20060	0.02330
4	3	1	1	480.	1.39600	0.20060
4	3	3	1	400.	0.77730	0.08840
5	2	1	1	960.	0.08840	0.00600
6	1	1	1	50.	2.64000	0.34740
6	2	2	1	960.	0.93970	0.08840
6	2	2	3	320.	0.54750	0.05130
6	2	3	2	400.	0.77730	0.05130
6	3	1	1	1040.	0.20060	0.02330
7	2	1	1	400.	0.54750	0.05130
7	2	3	1	560.	0.34740	0.02330
8	1	1	1	50.	0.54750	0.08840
8	2	2	1	50.	0.08840	0.02330
8	2	3	2	2080.	0.02330	0.02330
8	3	1	1	800.	0.08840	0.02330
8	3	3	1	880.	0.05130	0.00600

TABLE C.15 (Continued)

9	2	1	1	2240.	1.39600	0.20060
9	2	2	2	1760.	0.54750	0.05130
9	2	3	1	400.	0.54750	0.08840
9	2	3	3	100.	0.34740	0.05130
9	3	2	1	1200.	0.54750	0.08840
9	3	2	3	240.	0.34740	0.05130
9	3	3	2	2240.	0.20060	0.02330
10	1	1	1	480.	2.18200	0.34740
10	3	1	1	2080.	0.77730	0.08840
10	3	3	1	800.	0.54750	0.05130
10	3	3	3	480.	0.20060	0.02330
10	4	2	1	800.	0.34740	0.05130
10	5	1	1	1200.	0.34740	0.05130
11	2	1	1	1600.	0.93970	0.20060
11	3	2	1	720.	0.54750	0.05130
11	3	3	2	800.	0.20060	0.02330
11	3	4	1	800.	0.20060	0.02330
11	5	1	1	1200.	0.54750	0.05130
12	2	1	1	240.	0.20060	0.02330
12	3	2	1	2320.	0.77730	0.08840
12	3	2	3	1200.	0.34740	0.05130
13	1	1	1	160.	3.14200	0.34740
13	2	1	1	1920.	2.64000	0.34740
13	3	3	1	1680.	1.39600	0.20060
13	3	3	3	2240.	0.77730	0.08840

TABLE C.16

80% TUS PIPE DATA

I	J	K	L	PIPE LENGTH	PIPE AREA (1)	PIPE AREA (2)
1	1	1	1	16000.	0.93970	0.93970
1	2	2	1	400.	0.08840	0.08840
1	2	4	1	800.	0.20060	0.20060
1	4	1	1	8200.	0.34740	0.34740
1	4	2	1	3600.	0.20060	0.20060
1	4	3	3	3200.	0.20060	0.20060
1	4	5	2	80.	0.20060	0.20060
1	4	5	4	2320.	0.00600	0.00600
1	5	2	1	720.	0.02330	0.02330
2	1	1	1	50.	2.18200	0.20060
2	3	1	1	960.	1.39600	0.08840
2	3	2	2	800.	0.77730	0.08840
2	3	3	1	50.	0.08840	0.02330
3	1	1	1	400.	3.14200	0.34740
3	2	2	1	50.	0.08840	0.02330
3	3	1	1	1200.	2.18200	0.20060
3	3	2	2	560.	0.54750	0.05130
3	3	3	1	800.	1.39600	0.08840
4	1	1	1	50.	1.76700	0.20060
4	2	2	1	1440.	0.20060	0.02330
4	3	1	1	480.	1.39600	0.20060
4	3	3	1	400.	0.77730	0.08840
5	2	1	1	960.	0.08840	0.00600
6	1	1	1	50.	2.64000	0.34740
6	2	2	1	960.	0.93970	0.08840
6	2	2	3	320.	0.54750	0.05130
6	2	3	2	400.	0.77730	0.05130
6	3	1	1	1040.	0.20060	0.02330
7	2	1	1	400.	0.54750	0.05130
7	2	3	1	560.	0.34740	0.02330
8	1	1	1	50.	0.54750	0.08840
8	2	2	1	50.	0.08840	0.02330
8	2	3	2	2080.	0.02330	0.02330
8	3	1	1	800.	0.08840	0.02330
8	3	3	1	880.	0.05130	0.00600
9	2	1	1	2240.	1.39600	0.20060
9	2	2	2	1760.	0.54750	0.05130
9	2	3	1	400.	0.54750	0.08840
9	2	3	3	100.	0.34740	0.05130
9	3	2	1	1200.	0.54750	0.08840
9	3	2	3	240.	0.34740	0.05130
9	3	3	2	2240.	0.20060	0.02330
10	1	1	1	480.	2.18200	0.34740
10	3	1	1	2080.	0.77730	0.08840
10	3	3	1	900.	0.54750	0.05130
10	3	3	3	480.	0.20060	0.02330
10	4	2	1	800.	0.34740	0.05130
10	5	1	1	1200.	0.34740	0.05130
13	2	1	1	480.	0.77730	0.08840
13	3	2	1	480.	0.93970	0.20060
13	3	3	2	50.	0.77730	0.08840

Notes:

(1) Pipes sized for absorptive air conditioning

(2) Pipes sized for winter peak load

TABLE C.16 (Continued)

I	J	K	L	PIPE LENGTH	PIPE AREA	PIPE AREA
1	2	1	1	4640.	0.54750	0.54750
1	2	3	1	8320.	0.20060	0.20060
1	3	1	1	50.	0.20060	0.20060
1	4	2	1	80.	0.05130	0.05130
1	4	3	2	80.	0.05130	0.05130
1	4	5	1	1200.	0.20060	0.20060
1	4	5	3	640.	0.00600	0.00600
1	5	1	1	2320.	0.20060	0.20060
1	5	3	1	2000.	0.20060	0.20060
2	2	1	1	1200.	0.34740	0.02330
2	3	2	1	1440.	1.39600	0.08840
2	3	2	3	50.	0.34740	0.02330
2	4	1	1	2400.	0.54750	0.05130
3	2	1	1	1000.	0.34740	0.02330
3	2	3	1	720.	0.20060	0.02330
3	3	2	1	1680.	0.77730	0.08840
3	3	2	3	960.	0.34740	0.02330
3	4	1	1	3440.	0.93970	0.08840
4	2	1	1	320.	0.54750	0.05130
4	2	3	1	480.	0.34740	0.05130
4	3	2	1	1760.	0.54750	0.05130
5	1	1	1	50.	0.34740	0.02330
5	3	1	1	800.	0.20060	0.02330
6	2	1	1	160.	2.64000	0.20060
6	2	2	2	880.	0.54750	0.05130
6	2	3	1	400.	1.39600	0.20060
6	2	3	3	1600.	0.77730	0.08840
7	1	1	1	320.	1.39600	0.20060
7	2	2	1	800.	0.34740	0.02330
7	3	1	1	1200.	0.77730	0.05130
8	2	1	1	960.	0.54750	0.08840
8	2	3	1	2240.	0.54750	0.08840
8	2	3	3	400.	0.54750	0.08840
8	3	2	1	800.	0.02330	0.00600
9	1	1	1	480.	1.76700	0.34740
9	2	2	1	240.	0.77730	0.08840
9	2	2	3	1200.	0.34740	0.05130
9	2	3	2	1280.	0.20060	0.02330
9	3	1	1	400.	0.77730	0.20060
9	3	2	2	1120.	0.34740	0.05130
9	3	3	1	560.	0.34740	0.05130
9	3	3	3	320.	0.20060	0.02330
10	2	1	1	1200.	0.54750	0.08840
10	3	2	1	160.	0.20060	0.02330
10	3	3	2	1200.	0.34740	0.05130
10	4	1	1	1120.	0.54750	0.08840
10	4	3	1	1520.	0.20060	0.05130
13	1	1	1	160.	3.14200	0.34740
13	3	1	1	1920.	2.64000	0.34740
13	3	3	1	1680.	1.39600	0.20060
13	3	3	3	2240.	0.77730	0.08840

TABLE C.17

60% TUS PIPE DATA

I	J	K	L	PIPE LENGTH	PIPE AREA (1)	PIPE AREA (2)
1	1	1	1	16000.	0.77730	0.77730
1	2	2	1	400.	0.08840	0.08840
1	2	4	1	800.	0.20060	0.20060
1	4	1	1	8200.	0.20060	0.20060
1	4	5	1	1200.	0.20060	0.20060
1	4	5	3	640.	0.00600	0.00600
1	5	1	1	2320.	0.20060	0.20060
1	5	3	1	2000.	0.20060	0.20060
2	2	1	1	1200.	0.34740	0.02330
2	3	2	1	1440.	1.39600	0.08840
2	3	2	3	50.	0.34740	0.02330
2	4	1	1	2400.	0.54750	0.05130
3	2	1	1	1000.	0.34740	0.02330
3	2	3	1	720.	0.20060	0.02330
3	3	2	1	1680.	0.77730	0.08840
3	3	2	3	960.	0.34740	0.02330
3	4	1	1	3440.	0.93970	0.08840
4	2	1	1	320.	0.54750	0.05130
4	2	3	1	480.	0.34740	0.05130
4	3	2	1	1760.	0.54750	0.05130
5	1	1	1	50.	0.34740	0.02330
5	3	1	1	800.	0.20060	0.02330
6	2	1	1	160.	2.64000	0.20060
6	2	2	2	880.	0.54750	0.05130
6	2	3	1	400.	1.39600	0.20060
6	2	3	3	1600.	0.77730	0.08840
7	1	1	1	320.	1.39600	0.20060
7	2	2	1	800.	0.34740	0.02330
7	3	1	1	1200.	0.77730	0.05130
10	2	1	1	1200.	0.54750	0.08840
10	3	2	1	160.	0.20060	0.02330
10	3	3	2	1200.	0.34740	0.05130
10	4	1	1	1120.	0.54750	0.08840
10	4	3	1	1520.	0.20060	0.05130
13	1	1	1	160.	3.14200	0.34740
13	3	1	1	1920.	2.64000	0.34740
13	3	3	1	1680.	1.39600	0.20060
13	3	3	3	2240.	0.77730	0.08840

Notes:

(1) Pipes sized for absorptive air conditioning

(2) Pipes sized for winter peak load

TABLE C.17 (Continued)

I	J	K	L	PIPE LENGTH	PIPE AREA	PIPE AREA
1	2	1	1	4640.	0.54750	0.54750
1	2	3	1	8320.	0.20060	0.20060
1	3	1	1	50.	0.20060	0.20060
1	4	2	1	80.	0.05130	0.05130
1	4	5	2	80.	0.20060	0.20060
1	4	5	4	2320.	0.00600	0.00600
1	5	2	1	720.	0.02330	0.02330
2	1	1	1	50.	2.18200	0.20060
2	3	1	1	960.	1.39600	0.08840
2	3	2	2	800.	0.77730	0.08840
2	3	3	1	50.	0.08840	0.02330
3	1	1	1	400.	3.14200	0.34740
3	2	2	1	50.	0.08840	0.02330
3	3	1	1	1200.	2.18200	0.20060
3	3	2	2	560.	0.54750	0.05130
3	3	3	1	800.	1.39600	0.08840
4	1	1	1	50.	1.76700	0.20060
4	2	2	1	1440.	0.20060	0.02330
4	3	1	1	480.	1.39600	0.20060
4	3	3	1	400.	0.77730	0.08840
5	2	1	1	960.	0.08840	0.00600
6	1	1	1	50.	2.64000	0.34740
6	2	2	1	960.	0.93970	0.08840
6	2	2	3	320.	0.54750	0.05130
6	2	3	2	400.	0.77730	0.05130
6	3	1	1	1040.	0.20060	0.02330
7	2	1	1	400.	0.54750	0.05130
7	2	3	1	560.	0.34740	0.02330
10	1	1	1	480.	2.18200	0.34740
10	3	1	1	2080.	0.77730	0.08840
10	3	3	1	800.	0.54750	0.05130
10	3	3	3	480.	0.20060	0.02330
10	4	2	1	800.	0.34740	0.05130
10	5	1	1	1200.	0.34740	0.05130
13	2	1	1	480.	0.77730	0.08840
13	3	2	1	480.	0.93970	0.20060
13	3	3	2	50.	0.77730	0.08840

TABLE C.18

40% TUS PIPE DATA

I	J	K	L	PIPE LENGTH	PIPE AREA (1)	PIPE AREA (2)
1	1	1	1	16000.	0.54750	0.54750
1	2	2	1	400.	0.08840	0.08840
1	2	4	1	800.	0.20060	0.20060
2	1	1	1	50.	2.18200	0.20060
2	3	1	1	960.	1.39600	0.08840
2	3	2	2	800.	0.77730	0.08840
2	3	3	1	50.	0.08840	0.02330
3	1	1	1	400.	3.14200	0.34740
3	2	2	1	50.	0.08840	0.02330
3	3	1	1	1200.	2.18200	0.20060
3	3	2	2	560.	0.54750	0.05130
3	3	3	1	800.	1.39600	0.08840
4	1	1	1	50.	1.76700	0.20060
4	2	2	1	1440.	0.20060	0.02330
4	3	1	1	480.	1.39600	0.20060
4	3	3	1	400.	0.77730	0.08840
13	2	1	1	480.	0.77730	0.08840
13	3	2	1	480.	0.93970	0.20060
13	3	3	2	50.	0.77730	0.08840
1	2	1	1	4640.	0.54750	0.54750
1	2	3	1	8320.	0.20060	0.20060
1	3	1	1	50.	0.20060	0.20060
2	2	1	1	1200.	0.34740	0.02330
2	3	2	1	1440.	1.39600	0.08840
2	3	2	3	50.	0.34740	0.02330
2	4	1	1	2400.	0.54750	0.05130
3	2	1	1	1000.	0.34740	0.02330
3	2	3	1	720.	0.20060	0.02330
3	3	2	1	1680.	0.77730	0.08840
3	3	2	3	960.	0.34740	0.02330
3	4	1	1	3440.	0.93970	0.08840
4	2	1	1	320.	0.54750	0.05130
4	2	3	1	480.	0.34740	0.05130
4	3	2	1	1760.	0.54750	0.05130
13	1	1	1	160.	3.14200	0.34740
13	3	1	1	1920.	2.64000	0.34740
13	3	3	1	1680.	1.39600	0.20060
13	3	3	3	2240.	0.77730	0.08840

Notes:

(1) Pipes sized for absorptive air conditioning

(2) Pipes sized for winter peak load

TABLE C.19

20% TUS PIPE DATA

I	J	K	L	PIPE LENGTH	PIPE AREA (1)	PIPE AREA (2)
1	1	1	1	16000.	0.34740	0.34740
1	2	2	1	400.	0.08840	0.08840
1	3	1	1	50.	0.20060	0.20060
2	2	1	1	1200.	0.34740	0.02330
2	3	2	1	1440.	1.39600	0.08840
2	3	2	3	50.	0.34740	0.02330
2	4	1	1	2400.	0.54750	0.05130
3	2	1	1	1000.	0.34740	0.02330
3	2	3	1	720.	0.20060	0.02330
3	3	2	1	1680.	0.77730	0.08840
3	3	2	3	960.	0.34740	0.02330
3	4	1	1	3440.	0.93970	0.08840
4	2	1	1	320.	0.54750	0.05130
4	2	3	1	480.	0.34740	0.05130
4	3	2	1	1760.	0.54750	0.05130
1	2	1	1	4640.	0.34740	0.20060
1	2	4	1	800.	0.20060	0.20060
2	1	1	1	50.	2.18200	0.20060
2	3	1	1	960.	1.39600	0.08840
2	3	2	2	800.	0.77730	0.08840
2	3	3	1	50.	0.08840	0.02330
3	1	1	1	400.	3.14200	0.34740
3	2	2	1	50.	0.08840	0.02330
3	3	1	1	1200.	2.18200	0.20060
3	3	2	2	560.	0.54750	0.05130
3	3	3	1	800.	1.39600	0.08840
4	1	1	1	50.	1.76700	0.20060
4	2	2	1	1440.	0.20060	0.02330
4	3	1	1	480.	1.39600	0.20060
4	3	3	1	400.	0.77730	0.08840

Notes:
(1) Pipes sized for absorptive air conditioning
(2) Pipes sized for winter peak load

APPENDIX C.2

CONSUMER HEAT EXCHANGER SIZING

Consumer heat exchanger end-use equipment is considered to consist of a central low-temperature low-pressure heat exchanger, supplying heat to a building's heating system.

Conditions on the heat exchanger are:

TUS water supply temperature:	200°F
TUS water return temperature:	80°F
Consumer outlet temperature:	150°F
Consumer return temperature:	68°F
Average heat transfer coefficient:	$U = \frac{100 \text{ Btu}}{\text{hr}\cdot\text{ft}^2\,°\text{F}}$

Substituting these into the standard heat exchanger equation

$$Q = UALMTD, \qquad (C.2.1)$$

where

Q = thermal load delivered (Btu/hr)

A = heat transfer area (ft^2)

LMTD = log mean temperature difference

$$= \frac{(200° - 150°) - (80° - 68°)}{\ln \frac{200° - 150°}{80° - 68°}}$$

$$= 26.6°F$$

Rearranging Equation C.2.1 to solve for heat transfer area gives Eq. C.2.2:

$$A = \frac{Q}{2660} \qquad (C.2.2)$$

which is used to find the required consumer heat transfer area. Total heat transfer area is scaled in proportion to total base thermal demand.

References

1. "Mechanical Standards of TEMA Class 'C' Heat Exchangers," Chapter 6, Standards of the Tubular Exchanger Manufacturers Association, Fifth Edition, Tubular Exchanger Manufacturers Association, New York, 1968.

2. El-Wakil, M., Nuclear Heat Transport, International Textbook Compnay, Scranton, Penn., 1971, p. 243.

3. Goldman, S.G., Best, F.R., Golay, M.W., "TDIST2, A Computer Program for Community Energy Consumption Analysis and Total Energy System Design," Project Report, Contract No. DAAK02-74-C-0308, Department of Nuclear Engineering, MIT, June 1977.

APPENDIX D.1

CALCULATION OF HTGR CAPITAL COST

Metcalfe, et al., [1] calculates the costs of HTGR power plants using the CONCEPT III Code [1], arriving at the results shown in Fig. D.1 reproduced from his report [1]. This figure shows the variation of capital cost (in terms of dollars per Kwhr) versus power plant electrical capacity. These data are well-represented by the equation

$$\text{Unit Capacity Cost} = 16650.(\text{Mw(e)})^{-.497} \qquad \text{D.1}$$

where Unit Capacity Cost is stated in terms of 1985 dollars per KW(e), and the quantity - Mw(e) - refers to plant electrical capacity stated in Mw(e).

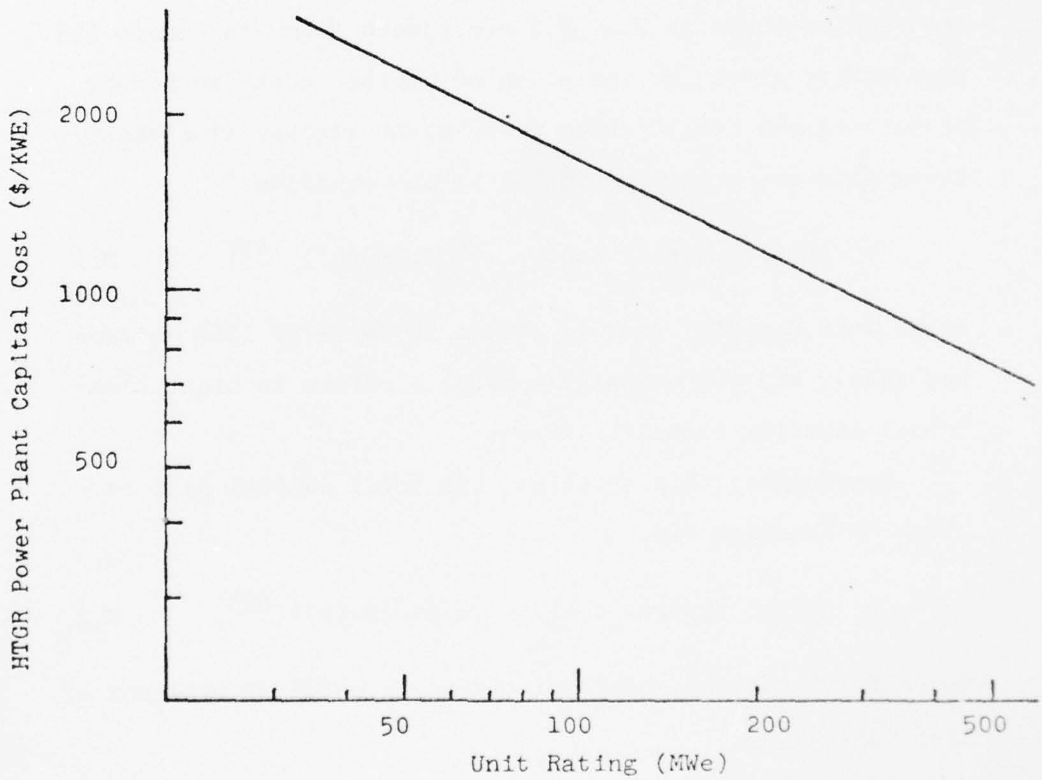
Rearranging this equation, the total capital cost is given by Equation D.2,

$$\text{Total Capital Costs} = 16.65\ (\text{Mw(e)})^{.503}, \qquad \text{D.2}$$

where Total Capital Costs are stated in units of millions of 1985 dollars.

The capital cost calculations are based on the assumptions listed in Appendix D.4.

Fuel and O/M costs are based on a cost of 28.5 mills/KW(e)hr. This value is taken from Metcalfe's work [1] and accounts for plant size and capacity factor.



Note: All costs are given in 1985 dollars.

Figure D.1. Unit Capital Cost as a Function of Rating

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ANALYSIS OF NUCLEAR AND COAL FUELED TOTAL ENERGY SYSTEM OPTIONS--ETC(U)
JUN 77 F R BEST, S B GOLDMAN, M W GOLAY DAAK02-74-C-0308
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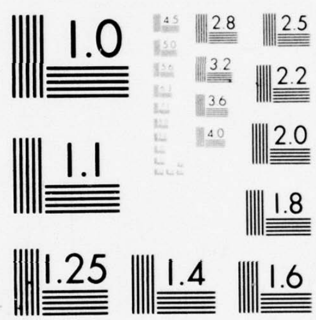


MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS-1963

APPENDIX D.2

CALCULATION OF COAL CONSUMPTION

Annual coal consumption for a given thermal/electric split is calculated by integrating the daily average coal consumption rate over the year. Coal consumption for a given day is found from the heat rate for the gas turbine generators and the electric and thermal loads calculated by TDIST2 (see Chapters 3, 4 and 5). Specifically the sequence of calculations is the following:

1. A twenty-four hour simulation of the Ft. Knox thermal and electrical power demands for a particular day at a given thermal electric load split is performed (Figure 5.34),
2. The gas consumption required for generation of the electrical energy demanded is calculated by using an average gas turbine heat rate of 13,500 BTU/KW(e)-hr [2],
3. The waste heat recovered from the turbine exhaust (as reported in the project Gas Turbine report [2]) is subtracted from the total thermal energy demand for the day. If the total thermal energy demanded exceeds the waste heat recovered from the turbine exhaust, additional gas is burned in a central hot water heater. (The extra gas which is burned in this fashion is assumed to supply energy at a rate of 5,000 BTU/KW(t)-hr),

4. The total amount of coal consumed during the day is found by adding gas consumption for electrical energy generation to any extra gas consumption for direct thermal heating and by converting gas consumption to coal consumption via an average gasifier coal-to-gas conversion efficiency of 70%,

5. The yearly coal consumption for a given thermal/electric load split is found by repeating steps 1 through 4 over the desired range of annual weather variation. This provides the basic data for the annual fuel consumption integration. In practice simulations for an average winter day, an average winter-spring day, an average spring-summer day and an average summer day are used in constructing an annual fuel consumption schedule . The annual fuel consumption data are then integrated over the year to obtain an estimate of the total annual fuel consumption rate.

Steps 1 through 5 must be repeated for each thermal/electric load split of interest. Additionally, the winter peak and summer peak design day simulations must be performed, since these days determine the TES maximum loads and load variations, and hence the required power generation and thermal reservoir equipment capacities. Samples of these calcula-

tions are the following:

1. Typical TDIST2 results for an average winter day, 80% thermal/electrical load split absorptive air conditioning option are

Average Thermal Demand	49.5 MW(t), and
Average Electric Demand	19.3 MW(e);

2. Using a gas turbine heat rate of 13,500 BTU/KWhr, the day's electrical generation gas consumption would be given as

$$\text{Gas Consumption for Electrical Generation} = 19.3\times10^3 \text{ KW(t)} \times 24 \text{ hrs} \; 13.5\times10^3 \text{ BTU/KWhr, or}$$

$$\text{Gas Consumption for Electrical Generation} = 6.25\times10^9 \text{ BTU of gas;}$$

3. The waste heat recovered from the generation of this electrical energy (from Ref. 2) would be determined as

$$\dot{Q} \text{ waste heat exchanger} = 653 \text{ MW(t)hrs,}$$

thus,

$$\text{Integral Thermal Demand} = 49.5\times10^3 \text{ KW} \times 24 \text{ hrs, and}$$

$$\text{Integral Thermal Demand} = 1188 \text{ MW(t)hrs}$$

$$\dot{Q}_{\text{waste heat exchanger}} = -653 \text{ MW(t)hrs}$$

$$\text{Extra heating gas burn} = 535 \text{ MW(t)hrs}$$

(using a heat rate of 5,000 BTU/KWhr), the extra heating gas burn requires production of

$$535 \text{ MW(t)hrs} \times 5{,}000 \text{ BTU/KWhr} = 2.68\times10^9 \text{ BTU;}$$

4. The total gas consumed for the day is the sum of electrical and heating gas consumption:

$$\text{Total Gas Consumption} = \text{Electrical + Heating Gas Consumption}$$

$$\text{Total Gas Consumption} = 6.25\times10^{9}\ \text{BTU} + 2.68\times10^{9}\ \text{BTU}$$

$$\text{or Total Gas Consumption} = 8.93\times10^{9}\ \text{BTU}.$$

For a typical gasifier efficiency of 70%, this requires a coal consumption given as

$$\text{Coal Consumption} = \frac{8.93\times10^{9}\ \text{BTU}}{.7}\text{, or}$$

$$\text{Coal Consumption} = 1.28\times10^{10}\ \text{BTU;}$$

5. Steps 1 through 4 are repeated for the other days of interest for the 80% thermal split TUS (and other splits of interest).

These daily consumption data are then integrated over the course of the year to get total annual coal consumption.

REFERENCES

1. Metcalfe, L.J., Driscoll, M.J., "Economic Assessment of Nuclear and Fossil-Fired Energy Systems for DOD Installations," Project Report, Contract No, DAAK02-74-C-0308, Department of Nuclear Engineering, MIT, February 1975.

2. Boyd, W.C., Golay, M.W., "Economic and Technical Aspects of Coal Gasification for Use in Gas Turbine Operation," Project Report, Contract No, DAAK02-74-C-0308, Department of Nuclear Engineering, MIT, 1976.

3. Kelly, J., Golay, M.W., "Economic and Technical Aspects of Gas Turbine Power Stations in Total Energy Applications," Project Report, Contract No. DAAK02-74-C-0308, Department of Nuclear Engineering, MIT, 1976.

APPENDIX D.3

The ultimate and proximate analyses of the coal which is assumed in the study to be consumed is summarized in Table D.3.1.

TABLE D.3.1

ASSUMED COAL ANALYSES

Ultimate Analysis

Carbon	57.1%
Hydrogen	3.9%
Oxygen	8.3%
Nitrogen	.8%
Sulfur	4.5%

Proximate Analysis

Moisture	12.3%
Ash	13.3%
Heating Value	9,500 BTU/lb

APPENDIX D.4

ECONOMIC GROUNDRULES

The economic groundrules used in Estimating TES Costs Over-life are summarized in Table D.4.1.

TABLE D.4.1

ECONOMIC GROUNDRULES USED IN ESTIMATING TES COSTS OVER-LIFE

Plant Types - HTGR/Brayton cycle
CGGT direct cycle

Date of Operation - 1985

Cost of Money - 10%

Average escalation rate - 6.3%

30 year plant lifetime

Straight line debenture accounting

APPENDIX D.5

EQUIVALENT COST OF FOSSIL FUEL

An example of the calculation of the equivalent breakeven cost of an alternative fuel is presented in the following example:

Case - Coal Costs = \$27/Ton (in 1985)

Thermal/Electrical Load Split = 80% Compressive Option

Cost Break-even · Mass Annual Fuel = Annual Cost (Capital, Operational Maintenance, and Coal) to run the TES, or

Fuel Cost × (9.43 × 1.81×10^{5} tons) = \$119.1×10^{6}

→ Breakeven Fuel Cost = \$69.8/ton

APPENDIX D.6

PIPE AND TRENCH COST DATA

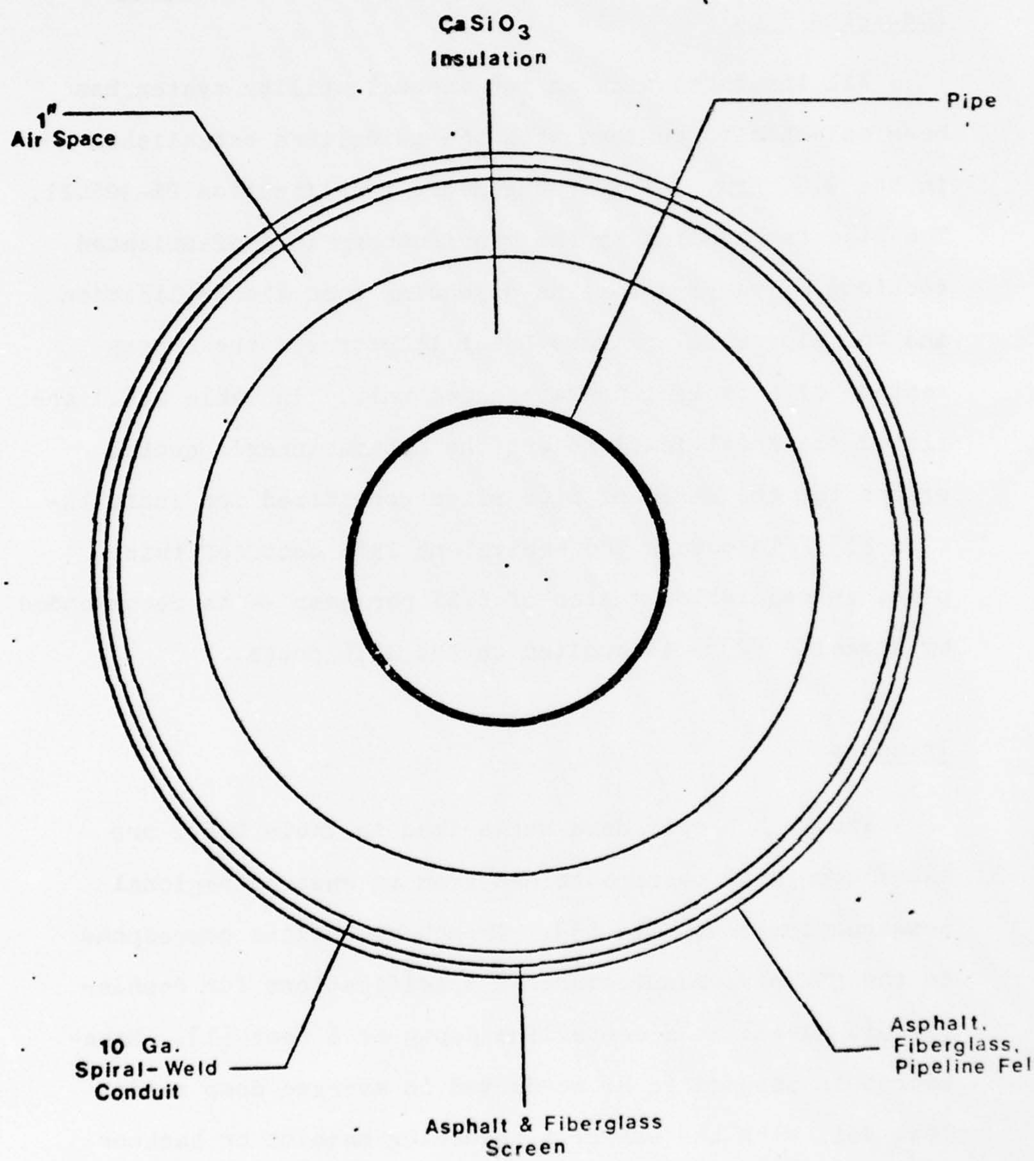
Insulated Pipe

All insulated pipe in the thermal utility system has been selected to conform with the guidelines established in the U.S. Army Corps of Engineers Specification CE-301.21. The pipe is supplied by the manufacturer in prefabricated sections of varying lengths depending upon the application and the pipe size. Figure D.6.1 illustrates the cross-section of a typical prefabricated unit. In Table D.6.1 are listed the specifications and the manufacturer's quoted prices for the range of pipe sizes considered for installation [1]. To obtain the equivalent 1985 costs of this pipe, an escalation factor of 6.2% per year — as recommended by Metcalfe [2] — is applied to the 1976 costs.

Trenches

The trench cost data summarized in Table D.6.2 are based upon unit costs obtained from an eastern regional construction cost file [3]. Trench dimensions correspond to the HTW pipe manufacturer's specifications for double-circuit burial at a centerline depth of 6 feet [1]. Excavation is assumed to be conducted in average damp sandy loam soil with the use of a trenching machine or backhoe.

Figure D.6.1
Cross-Section of Prefabricated HTW Transmission Pipe



(from Ref. 1)

Backfilling is by bulldozer or backhoe from fill deposited at the trench edge, and the backfilled soil is compacted with an air-powered tamping machine. 1985 costs are obtained by escalating the 1976 data at 6.2% per year.

TABLE D.6.1

PREFABRICATED INSULATED PIPE COSTS

Pipe O.D. (inches)	Pipe Wall Specification	Insulation Thickness (inches)	Jacket O.D. (inches)	1976 Cost ($/ft)	1985 Cost[(1)] ($/ft)
2	Sched. 40	1-1/2	8-5/8	18	31
3	Sched. 40	2	10-3/4	25	43
4	Sched. 40	2	10-3/4	26	45
6	Sched. 40	2-1/2	14	38	65
8	Sched. 40	2-1/2	16	46	79
10	Sched. 40	2-1/2	19	58	100
12	.375 wall	3	22	81	139
18	.375 wall	4	30	132	227
24	.375 wall	4	36	172	296

(1) Escalated at 6.2% per year from 1976.

TABLE D.6.2

TRENCHING COSTS

Pipe O.D. (inches)	Trench Dimensions (feet)	Excavation Cost(1) ($/lin.ft.)	Backfill Cost(2) ($/lin.ft.)	1976 Total Cost ($/lin.ft.)	1985 Total Cost ($/lin.ft.)
2	4 x 6-1/2	1.54	2.94	4.48	7.69
3	4 x 6-1/2	1.54	2.94	4.48	7.69
4	4 x 6-1/2	1.54	2.94	4.48	7.69
6	4 x 6-1/2	1.54	2.94	4.48	7.69
8	4 x 7	1.66	3.16	4.82	8.29
10	4-1/2 x 7	1.87	3.56	5.43	9.33
12	5 x 7	2.07	3.95	6.02	10.35
18	6-1/2 x 7-1/2	2.89	5.51	8.40	14.43
24	7-1/2 x 7-1/2	3.33	6.35	9.68	16.64

(1)Base cost = $1.60/cubic yard [5].

(2)Base cost = $3.05/cubic yard [5].

(3)1976 costs escalated at 6.2% per year.

REFERENCES

1. Personal communication, representative of Kenyon-Barstow Co., Division of Ric-Wil, Inc.

2. Metcalfe, L.J., "Economic Assessment of Alternative Total Energy Systems for Large Military Installations," MIT Department of Nuclear Engineering, S.M. Thesis, August 1976.

3. Building Construction Cost Data 1976, Robert S. Means Co., Inc., Duxbury, Mass., 1976.

APPENDIX D.7

PUMPING POWER COSTS AND PUMP RATING CALCULATIONS

Pumping Power

The pumping power required to overcome a given fluid frictional pressure loss is given by Eq. (D.7.1).

$$W = \frac{\Delta P A_c V}{737.56} = \frac{\Delta P \dot{m}}{737.56\rho} \qquad \text{(D.7.1)}$$

where

W = pumping power (kW),

ΔP = fluid pressure drop (lbf/ft^2),

A_c = flow channel cross section (ft^2),

V = fluid velocity (ft/sec),

$\dot{m}$ = fluid mass flowrate (lbm/sec),

ρ = fluid density (lbm/ft^3), and

1kW = 737.56 ft-lbf/sec.

Thus, knowing the fluid mass flowrate and the pipe dimensions for each loop, the Darcy pressure drop formula (Eq. (D.7.2)) may be used to compute the fluid frictional pressure losses, which are used in Eq. (D.7.1) to determine the pumping power requirements for the loop.

$$\Delta P = f \frac{L}{D} \frac{\rho V^2}{2g_c} \qquad \text{(D.7.2)}$$

where

ΔP = fluid pressure drop (lbf/ft^2),

L = flow channel length (ft),

D = flow channel diamter (ft),

ρ = fluid density (lbm/ft^3),

V = fluid velocity (ft/hr),

g_c = conversion factor = 4.17×10^8 $lbm\text{-}ft/lbf\text{-}hr^2$,

f = Darch-Weisbach friction factor, and

$$f \simeq \frac{0.184}{Re^{0.2}} ,$$

where Re is the fluid Reynolds number for turbulent flow.

Pumping Power Costs

TDIST2 uses a form of Eq. D.7.1 to calculate the pumping power required for each loop in the TUS at each time step. This pumping power is converted to electrical demand by assuming a 60% electrical-mechanical pump efficiency. This electrical demand is then added to the base total electrical demand, and is reflected in fuel consumption, and thereby throughout the 30 year life of the system.

Pump Rating and Costs

Although the average utility system fluid flowrates are determined primarily by the thermal energy demands experienced during the spring and fall months, the pumps must be sized to supply the peak system design conditions, and they operate at relatively low capacity factors throughout most of the year. Equation (D.7.3) can be used to

convert the design fluid mass flowrates from units of pounds per hour to units of gallons per minute, which can be used directly in the centrifugal pump cost function shown in Fig. D.7.1, adapted from the work of Ayorinde [1].

$$GPM = \frac{7.48\dot{m}}{60\rho} \tag{D.7.3}$$

where

GPM = fluid volume flowrate (gal/min),

$\dot{m}$ = fluid mass flowrate (lbm/hr),

ρ = fluid density (lbm/ft^3),

1 hr = 60 min, and

1 ft^3 = 7.48 gal.

Although the costs in Ayorinde's work are presented in 1973 dollars, the cost function shown in Fig. D.7.1 has been escalated at 6.2% per year — following the work of Metcalfe [2] — to obtain equivalent 1985 pump costs. Due to excessive pump component loading, the maximum pump rating recommended for general applications is 3000-3500 gpm [1]. In cases requiring ratings larger than this limit, it is assumed that two or more units are installed to divide the load equally.

Using these criteria, Table D.7.1 lists pump location, capacity, and cost for each TUS option studied. Recall from the discussion of Section 6.4.1 that secondary loops are relatively independent of one another with regard to pipe and pump sizing. Therefore Table D.7.1 lists each

secondary loop only once for each option studied, either compressive air conditioning or absorptive air conditioning. However, the primary loop for each TUS option and thermal/ electric split value is unique, hence primary loops are listed according to thermal/electric split value.

TABLE D.7.1

TUS PUMP RATINGS AND COSTS(1)

PUMP LOCATION I	J	K	L	CAPACITY (millions of lb/hr)	COST (millions of 1985 dollars)
A. 100% TUS, Absorptive Air Conditioning Option:					
1	1	1	1	1.75	.023
1	4	1	1	.84	.014
1	2	3	1	.27	.006
1	4	4	1	.31	.007
1	4	5	1	.20	.005
B. 80% TUS, Absorptive Air Conditioning Option:					
1	1	1	1	1.40	.018
1	2	3	1	.27	.006
1	4	1	1	.52	.009
1	4	3	1	.25	.006
1	4	5	1	.20	.005
1	4	5	4	.003	.0003
1	4	5	3	.008	.0005
1	4	1	1	.519	.009
1	1	1	1	1.40	.018

TABLE D.7.1 (Continued)

PUMP LOCATION I	J	K	L	CAPACITY (millions of lb/hr)	COST (millions of 1985 dollars)
C. 60% TUS Absorptive Air Conditioning Option:					
1	1	1	1	1.18	.017
1	2	3	1	.27	.006
1	4	1	1	.27	.006
1	4	5	1	.20	.005
1	4	5	4	.004	.0004
1	4	5	3	.007	.0006
1	4	1	1	.27	.006
1	1	1	1	1.18	.017
1	2	3	1	.27	.006
D. 40% TUS Compressive Air Conditioning Option:					
1	1	1	1	.73	.001
1	2	3	1	.27	.0006
1	1	1	1	.73	.001
1	2	3	1	.27	.0006
E. 20% TUS Absorptive Air Conditioning Option:					
1	1	1	1	.46	.009
1	2	1	1	.31	.007
1	1	1	1	.46	.009
1	2	1	1	.31	.007

TABLE D.7.1 (Continued)

PUMP LOCATION I	J	K	L	CAPACITY (millions of lb/hr)	COST (millions of 1985 dollars)
F. Secondary Loops, Absorptive Air Conditioning Option:					
2	1	1	1	3.12	.031
3	1	1	1	5.31	.045
4	1	1	1	2.81	.028
5	1	1	1	.41	.008
6	1	1	1	4.02	.036
7	1	1	1	2.00	.023
8	1	1	1	.92	.014
9	1	1	1	2.95	.030
10	1	1	1	3.25	.032
11	1	1	1	1.48	.019
12	1	1	1	1.30	.017
13	1	1	1	4.89	.043
G. 100% TUS Compressive Air Conditioning Option:*					
1	1	1	1	1.79	.023
1	2	3	1	.28	.006
1	4	1	1	.86	.014
1	4	4	1	.32	.007
1	4	4	3	.08	.003
1	4	3	1	.26	.006
1	4	5	1	.20	.005

*Pumps are in supply and return lines.

TABLE D.7.1 (Continued)

PUMP LOCATION I	J	K	L	CAPACITY (millions of lb/hr)	COST (millions of 1985 dollars)
H. 80% TUS Compressive Air Conditioning Option:					
1	1	1	1	1.35	.019
1	2	3	1	.27	.008
1	4	1	1	.50	.010
1	4	3	1	.25	.006
1	4	5	1	.18	.005
1	4	5	4	.004	.0003
1	4	5	3	.008	.0005
1	4	1	1	.50	.009
1	1	1	1	1.35	.019
1	2	3	1	.27	.006
I. 60% TUS Compressive Air Conditioning Option:					
1	1	1	1	1.11	.017
1	2	3	1	.27	.006
1	4	1	1	.25	.006
1	4	5	1	.18	.005
1	4	5	4	.004	.0003
1	4	5	3	.007	.0006
1	4	1	1	.25	.006
1	1	1	1	1.11	.017
1	2	3	1	.27	.006

TABLE D.7.1 (Continued)

PUMP LOCATION I	J	K	L	CAPACITY (millions of lb/hr)	COST (millions of 1985 dollars)
J. 40% TUS Compressive Air Conditioning Option:					
1	1	1	1	.74	.013
1	2	3	1	.28	.006
1	1	1	1	.74	.013
1	2	3	1	.28	.006
K. 20% TUS Compressive Air Conditioning Option:					
1	1	1	1	.47	.0092
1	2	1	1	.31	.0071
1	1	1	1	.47	.0092
1	2	1	1	.31	.0071
L. Secondary Loops, Compressive Air Conditioning Option:*					
2	1	1	1	.232	.0054
3	1	1	1	.391	.0077
4	1	1	1	.304	.0065
5	1	1	1	.029	.0013
6	1	1	1	.353	.0072
7	1	1	1	.151	.0040
8	1	1	1	.148	.0040
9	1	1	1	.374	.0075
10	1	1	1	.378	.0075
11	1	1	1	.459	.0086
12	1	1	1	.167	.0043
13	1	1	1	.546	.0097

*Pumps are in supply and return lines

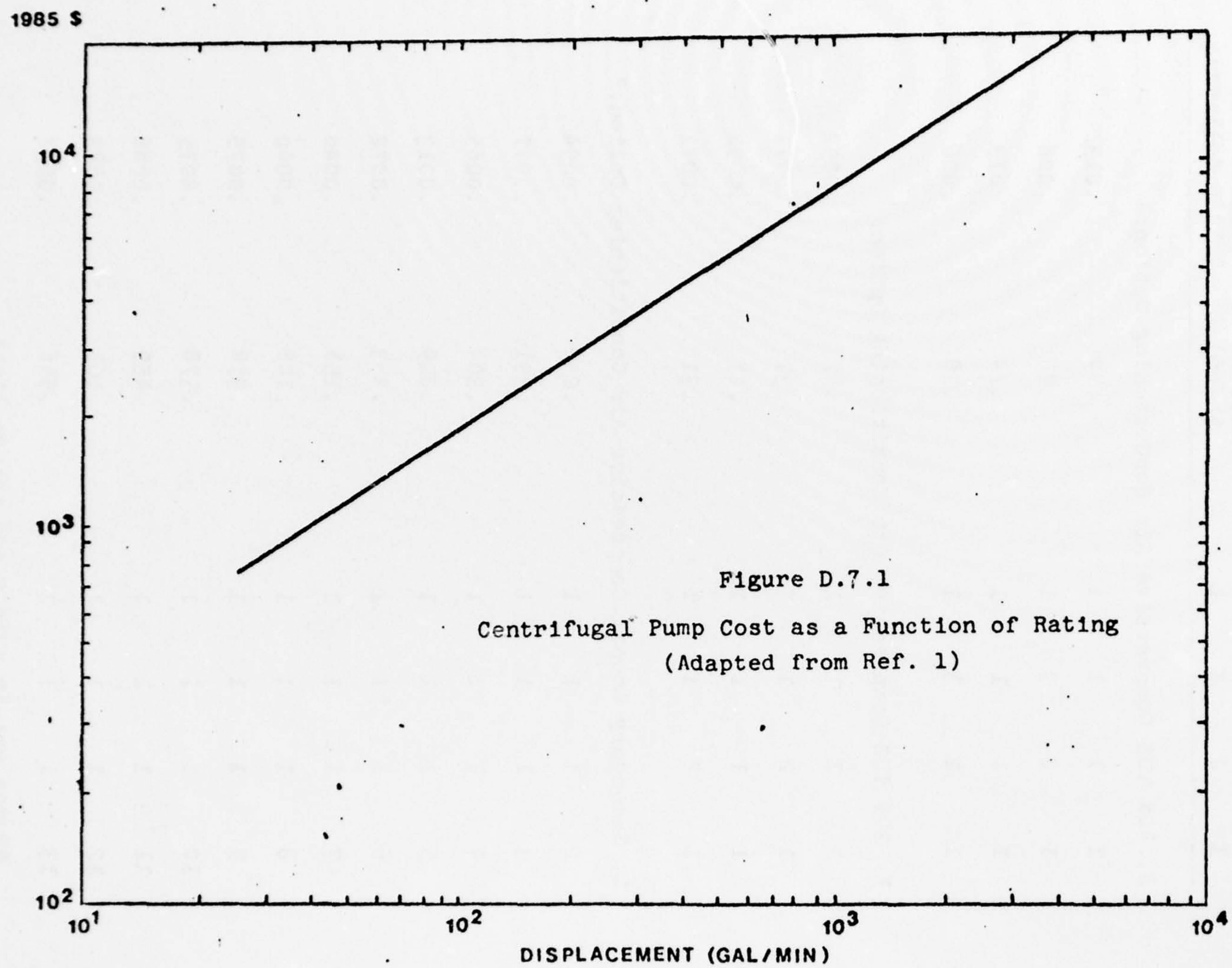


Figure D.7.1

Centrifugal Pump Cost as a Function of Rating

(Adapted from Ref. 1)

REFERENCES

1. Ayorinde, E.O., "Underground Transmission of Heat," MIT Department of Mechanical Engineering, S.M. Thesis, August 1973.

2. Metcalfe, L.J., "Economic Assessment of Alternative Total Energy Systems for Large Military Installations," MIT Department of Nuclear Engineering, S.M. Thesis, August 1975.

APPENDIX D.8

BUILDING CONNECTION PIPE COSTS

As mentioned in Section 6.4.1, the cost of the piping required to connect individual buildings to the water main in the street is an important, but not a dominating item in the total piping cost. The cost of the piping required to connect buildings to the street mains is calculated by correlating pipe length with Btu delivered. The actual length of pipe required to connect every building in the Van Voorhis and Dietz Acres developments to their nearest load center was measured. A pipe size of one inch nominal diameter was selected as a size suitable for individual home delivery. The total cost of this piping was found by multiplying the total length of pipe by the appropriate pipe cost. This total delivery pipe cost was then divided by the peak winter power supplied to Van Voorhis and Dietz Acres to obtain a cost for distribution piping on a peak power-delivered basis (in units of dollars per peak Btu per hour in 1985). This cost was then extrapolated to the entire base's distribution system by multiplying peak power delivery cost by the total base's peak energy delivery rate, giving the total cost for individual building distribution piping. It should be emphasized that this technique was used to determine only the cost of piping from the load centers to the buildings. The cost of piping in the TUS

is explicitly calculated by TDIST2 on a piece-wise basis. Finally, the cost of the distribution piping so calculated is seen always to be less than 20% of the total TUS cost, indicating that small errors in this cost component would introduce insignificant errors in the overall system cost estimate. Piping costs are shown in Table 6.5.

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